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ABSTRACT

This publication is a Mechanical Engineering Laboratory Manual designed to be used by technical institute students in Mechanical Engineering Technology Programs. The experiments are introductory in nature and embrace the fields of applied thermodynamics, fluid mechanics, refrigeration, heat transfer and basic instrumentation. There are 20 experiments in all. Each one begins with the necessary theoretical explanation followed by a list of references, objects, apparatus, procedures, and evaluation of results. Charts, diagrams, and results and data sheets to be completed are also included. (Author/GA)

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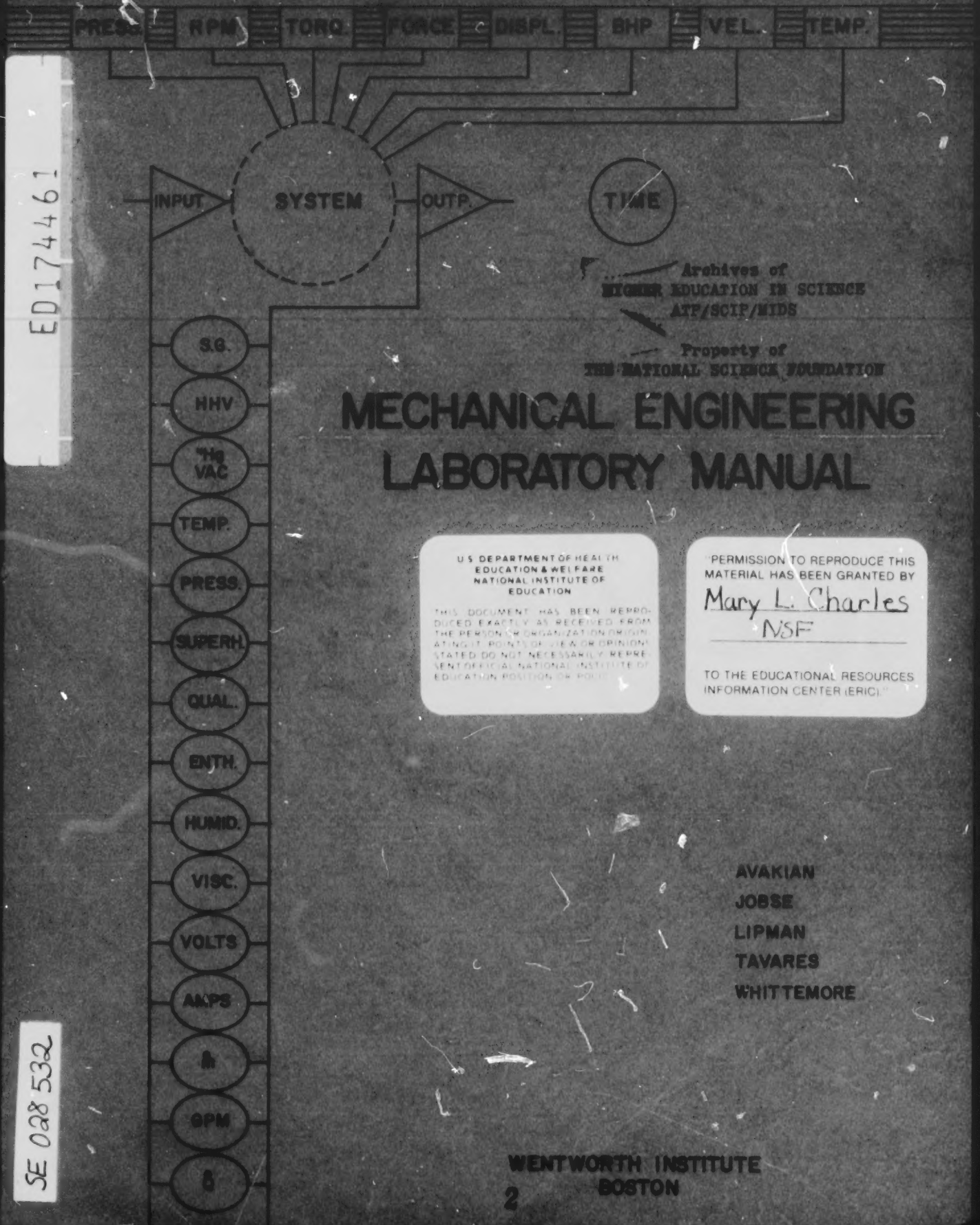

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MECHANICAL ENGINEERING LABORATORY MANUAL

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Mechanical Engineering Laboratory Manual

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This manual is designed to be used by technical institute students in Mechanical Engineering Technology Programs. The experiments are introductory in nature and embrace the fields of applied thermodynamics, fluid mechanics, refrigeration, heat transfer and basic instrumentation.

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A graph is a pictorial representation of data. It must fulfill a function that a simple tabulation of the data cannot; otherwise, the graph is a waste of time. A good graph is easy to understand, attractive in appearance, and, at a glance, brings home with impact the idea inherent in the data which the grapher wishes to display to the observer. A graph should be an accurate representation of the data, in that, it may be used as the primary source of data by the observer.

Good graphing requires the best talents of logic, accuracy and artistic temperament that an individual can master. It is not a science, in that, there is no one right way to picture the data. Many different representations of the same data may be equally effective in accomplishing a given purpose.

There are certain guidelines which have generally been accepted as basic to good graphing and incorporation of these ideas into your graphs is expected in the scientific community. Some of these guidelines have sound and practical reasons for being and are not arbitrary rules. Others however, are customs established to bring a uniformity to the art so that the observer can more easily get the intended message from the graph.

The following guidelines should be incorporated in all scientific graphical representations with rare exceptions.

Guidelines:

1. Use a good grade of graph paper. Quadruled paper is only for rough sketches and preliminary work.
2. Do not use the margin of the graph paper; it is used for appearance and building and during duplication will most likely, in part, be lost.
3. Use simple scales and titles (including units) and number them. Simple scales are easy to plot and easy to get data from.
4. The final graph should be done in ink for neatness, permanence and reproduction quality.

5. Assign a complete title to the graph and reference it to the work with which it is connected. You should also sign or initial and date the graph in small size, in some non-prominent part of the graph paper.

6. The graph in its entirety should be neat and accurate.

7. Size and scale the graph to give the most effective presentation.

8. Color may be used to differentiate sets of data, but color sometimes is a problem because in duplication of the graph the color is lost. Other techniques may be used for data differentiation, such as, dashed, dotted and solid lines.

9. Standard graph paper should be viewed only in one of two orientations. Either wide margin to the left, or above is conventional, depending on the need for height or width of the graph. All written matter and titles should be oriented in keeping with the normal viewing position of the graph.

10. Data plotted should be clearly shown as points, normally ennunciated with a circle, square, cross or other symbol about the point. When a graph does not show the data points, it is usually implied that the graph is an approximation and is only effective in revealing an idea and not quantitative information specifically. Also, since no points may be seen, the scattering of data about the curve can not be seen, and thus no value or worth may be placed on the data which is the basis for the curve.

11. Because of experimental or other errors, data may be a mixture of bad and good points when plotted. Bad data points usually show up as an impossible (theoretically) irregularity of curve points. If a point is offset from an otherwise uniform pattern of points, which cannot be justified as a possible trend in data variation shown by other points surrounding it, it is most likely a point in error. Points in error, although they should still be displayed on the graph, should not be used in the drawing of the curve or line which is to represent the story the data

is telling through the talents of the grapher. In many cases, a fair degree of judgment and knowledge of the data source must be exercised in the evaluation of points seemingly in error.

Generalized Procedure:

Assuming you have the data to be plotted and the paper to be used, the following generalized procedure for preparing the graph may be used. Some modification may be required by special cases or individual preference.

1. Examine the data; make a crude graph of it on quadruled paper and see what the data is trying to convey in meaning. In some cases the meaning to be conveyed is already known, and this step would be a verification of what you expected. In any case, a rough sketch is very important to good graphing.

2. Once you have decided what the data is to convey to the observer in graphical form, decide on the size of the graph, on the paper, and the scales to be used to bring out this message forcefully. Remember, the graph does not have to fill up the entire paper; you can orient the graph in two ways and scales must be kept simple. Sometimes making another scaled sketch to size on quadruled paper will help, otherwise you will be relying on your first crude sketch to guide your decisions.

3. Now, on your finished graph paper, lightly pencil in the scale lines and label them by a few lightly pencilled numbers. Then, plot your data lightly, but accurately, in pencil.

4. Look at your lightly pencilled, incomplete graph. Does it look promising? If so, continue; if not, try again!

5. At this time, you must have a judgment as to the form of the curve (linear, exponental, etc. . .) either from a knowledge of the theory behind the data or from general trend of the points as lightly plotted. Erratic, non trend points must be excluded from this judgment as being in error, unless they can be justified in some way.

6. Using a French curve and ruler or other devices, pencil in the best smooth curve which represents the data's message, bearing in mind that some points, although plotted, will be disregarded in the graphing of the curve. Also keep in mind that there is bound to be a certain amount of error in all your points if obtained experimentally, but you assume it is random and your curve should be the best approximation of all the points considered. The majority of points should fall close to your curve, either to one side or the other. Don't try to draw the curve through all the points.

7. Once the proper curve has been drawn (and this may take several trials), you are ready to darken in the graph and complete it. From here on in, art is important, so don't lose everything you've done to this point with sloppy printing and inkwork. Now you can ink in the curve, points of data (including the bad ones), scale lines, scale values (complete), and enunciate the data points with one of the methods previously described or your own method if it works.

8. Now we must do the printing work on the graph in ink. It is assumed that careful attention has been given to the selection of titles to preserve a clear and descriptive meaning. If you don't do free hand printing well, then you may wish to pencil in the words before inking. When everything is ready, ink in graph, titles, and references, as to the source of the graph if required, scale titles and units, and notes required for classification or identification of graph components or meaning. Be sure to keep printing size in proportion to the importance of the item being done. For example, a reference would be printed in smaller sized letters than the graph title.

9. At this point, you should erase any penciled guidelines that show and clean up the graph of smudges, etc. Then, inspect the graph to see that it is complete and free of errors.

10. The graph is now complete except for your signature or initials and date in ink. Find some unobtrusive, suitable space, usually a corner for this information. Don't use the borders and keep it small, but feel free to use any artistic flair you desire as long as you can be identified.

GRAPHING TECHNIQUE

OBJECT

To develop, from given sets of data, a uniform graphing technique consistent with standards acceptable by industry.

PROCEDURE

Prepare four different graphs from assigned data. To give a uniformity and basis for comparison between different works you will use the paper specified and the scale pre-chosen for each of the four graphs. Each set of data is described sufficiently well for you to select your own title and labeling requirements, if not specified.

Graph Number 1

The following relationship gives the millivolts output versus temperature in degrees Fahrenheit for an experimental thermocouple.

$E = 0.020T + 0.80$

E = Voltage in millivolts

T = Temperature in ^{o}F

Plot the graph of the relationship as follows:

1. Abscissa - temperature in ^{o}F, from 0 to $400^{o}F$ using a scale of 1 inch = $50^{o}F$.
2. Ordinate - Thermocouple output in millivolts (MV), from 0 to 9 MV using a scale of 1/2 inch = 1MV.
3. Use the following titles -
   a. Abscissa - Thermocouple Temperature in (^{o}F)
   b. Ordinate - Thermocouple Output (Millivolts)
   c. Graph Title - Experimental Thermocouple No. 409A
      Voltage Output Vs. Temperature
      (reference junction at $-40^{o}F$)
4. From your graph determine the slope of the curve (MV change per ^{o}F) and record this on the graph in a suitable

place on the paper clearly identified.

5. Sign or initial and date your graph appropriately.

Graph Number 2

In December of 1962 the Chrysler engineering staff had received an order to develope and engineer a vehicle that could win closed circuit stock car events throughout the country. The designing of the new engine started in January 1963; the day of Daytona. This was the birth of the 426 cubic inch hemispherical engine from Chrysler. For drag racing, a two four-barrel carb version, and for the ovals, a single four-barrel carbureted 426 engine. As was expected it won at Daytona and many other races throughout the remainder of 1964.

Some measurements of the engines performance are listed below using dual four-barrel carburetors for dragging.

ENGINE RPM	BRAKE HORSEPOWER	TORQUE
3200	-----	445
3600	-----	470
4000	-----	494
4400	435	518
4800	485	535
5200	524	530
5600	556	522
6000	580	512
6400	595	500
6800	600	465

Plot these values using Brake-Horsepower and Torque as ordinates and Engine RPM as abscissa. Use one sheet of graph paper and the designated scales.

a. Abscissa scale 1 inch = 800 rpm

b. Ordinate scales

   For Brake-Horsepower 1 inch = 40 Bhp

   For Torque 1 inch = 40 lbf-ft

c. Determine appropriate title or titles and labels for the curves and graphs.

d. Start graph at (3000, 400)

Graph Number 3

In the operation of a gas processing plant, every 80 seconds, in a uniform cycle, gas is released from a pressure system to a vacuum system with a resultant rise in the pressure of the vacuum system. The amount of gas released each cycle is a function of operating level of the plant; thus, the higher the operating level the greater is the use of pressure in the vacuum system during the cycle.

The danger is, that, if the pressure in the vacuum system becomes too high, the vacuum producing machines become overloaded and automatically shut-down because of overload protection on them. This shuts down the whole plant, of which, the vacuum system is a very small but critical part. The following data was taken during normal operation (average operating level of plant) to show the relationship of the pressure rise in the vacuum system to the pressure at which the shut-down occurs.

The following additional information is supplied:

Vacuum machine shut-down pressure = 5.0 psia

Pressure release to vacuum system - The word used to describe the pressure release to the vacuum system is "AUSPUFF" and auspuff is always initiated 10 seconds after the start of each 80 second cycle.

Use the following scales for your graph:

Cycle time - 10 seconds to the inch

Pressure - 2.0 psia to the inch

NOTE: Your graph must display to management (in a report), how close the plant is to shut-down due to the vacuum machine overload protection.

DATA (measured during normal plant operating conditions)

Cycle Time (Sec)	Vacuum System Pressure (psia)
0	1.2
5	1.1
10	1.0
15	1.2
20	2.4
25	3.9
30	4.6
35	4.8
40	4.6
45	4.1
50	3.4
55	2.7
60	2.1
65	1.7
70	1.5
75	1.3
80	1.2

Graph Number 4

An ideal gas undergoes a certain process, such that the process is described by the equation:

$$PV^n = C$$

Plot with pressure as the ordinate the data listed below on regular graph paper. (10 or 20 squares per inch).

Also plot the data on logarithmic paper 1 by 1 cycles. (Important)

Determine from the logarithmic plot, the slope (m). (Keep in mind that you are reading values from log paper). Prove that the exponent (n) of the above equation is equal to the negative value of the slope.

Determine the value of the constant (C) with units included ($PV^n=C$).

NOTE: Do not convert units in plotting the values. Plot as they are.

Data: (Taken while process was occuring)

P (psia)	V (in^3)
800	20
700	21.5
670	22
540	24.5
520	25
480	26
430	27.5
400	28.5
360	30
300	33
265	35
250	36
225	38
175	43
160	45
145	47
140	48
115	53
110	54

Pressure may be defined as a force per unit area. It is one of the most common measurements made in industry. A variety of instruments are used all generally working on the idea of detecting some physical characteristic change, such as, height or volume which results from the application of a force. These pressures can be determined in reference to any arbitrary datum. The two most common references used are conveniently atmospheric pressure and zero pressure (absolute vacuum).

The following are some of the more commonly used terminology:

1. Barometric pressure is the atmospheric pressure measured in reference to zero pressure.
2. Gage pressure is the pressure above atmospheric measured relative to atmospheric pressure.
3. Vacuum gage pressure is the pressure below atmospheric measured relative to atmospheric pressure.
4. Absolute pressure is the pressure relative to zero pressure.

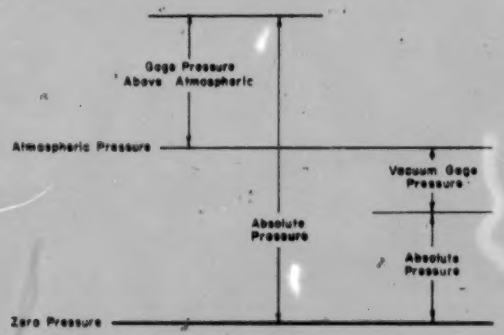

Figure 1. Pressure Measurement Terms

Figure 1 shows the relationship between gage and absolute pressures. If the gage reads above atmospheric, the absolute pressure is calculated by adding the gage pressure to the atmospheric pressure.

$$P_{abs} = P_{atm} + P_{gage}$$

If the gage reads vacuum (below atmospheric) the absolute pressure is calculated by subtracting the gage pressure from the atmospheric.

$$P_{abs} = P_{atm} - P_{gage}$$

The Bourdon tube is a compact and convenient instrument for measuring pressure. Its primary use is to measure high pressures up to a range of 100,000 psi. It can also be used to measure low pressures above or below atmospheric. It makes use of a flattened hollow metallic curved tube which is open at one end to accept the applied pressure and free at the other end. Application of pressure on the open end causes the tube to unwind. This movement of the free end is transmitted to a pointer which rotates over a calibrated scale giving a mechanical indication of pressure. The amount of movement depends on the difference in pressure across the wall of the Bourdon tube. Thus the pressures read are directly in reference to the surrounding atmospheric pressure and are gage readings. The scale may be calibrated to read in any convenient units, such as, pounds per square inch, pounds per square foot, inches of water, etc. The compound gage is a versatile Bourdon tube gage. It can be used to measure pressures, both above and below atmospheric.

Manometers are the basic or standard way of measuring pressure. They are excellent devices for measuring low pressures and may be used to measure gage pressure which is above atmospheric pressure or vacuum pressure which is below atmospheric pressure. The unknown pressure may be expressed directly as height of liquid Δh as shown by the U-tube manometer in Figure 2 or may be expressed in pounds per square inch by the following equation:

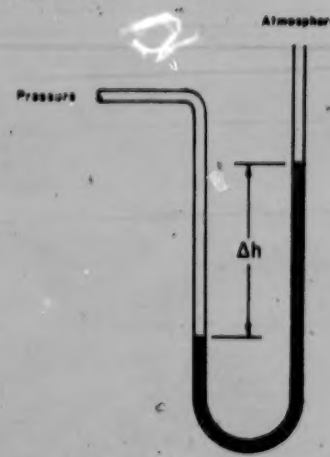

Figure 2. Simple Open U-tube Manometer

$$P = k\gamma\Delta h$$

γ = Specific weight of liquid in manometer in lbf/ft^3

$\gamma = \rho\, g/g_c$

ρ = density in lbm/ft^3

g = local acceleration of gravity in ft/sec^2

g_c = gravitational constant = 32.2 lbm-ft/lbf-sec^2

Δh = height of liquid in inches

K = conversion constant = $ft^3/1728\ in^3$

The well manometer (Figure 3) is a modification of the U-tube manometer. A slight change of height of mercury in the well results in a large Δh in the narrow leg. The zero of the scale is set at the level of the liquid in the well. This reference could be fixed or movable. If the scale is fixed, a correction factor may be necessary to account for the fluid level change in the well. The correction factor becomes negligible as the diameter of the well becomes much larger than the diameter of the tube (approximately in the order of 500 to 1).

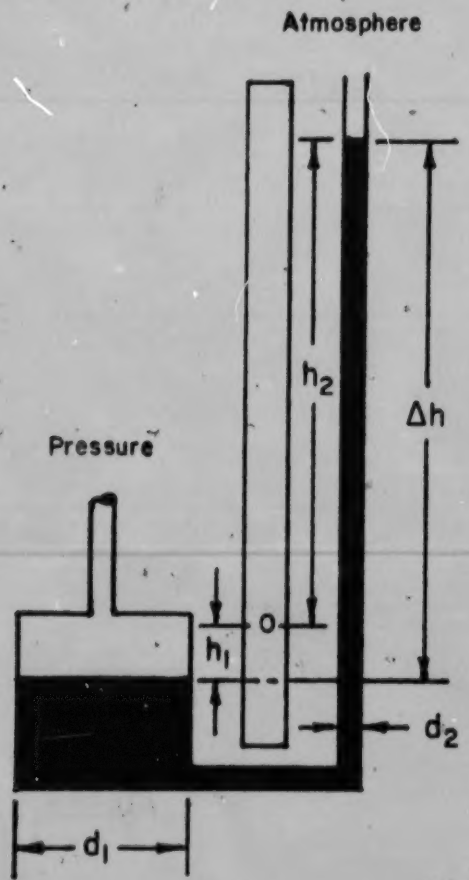

Figure 3 Well-Type Manometer

The pressure is calculated by the equation:

$$P = k\gamma\, \Delta h$$
$$= k\gamma\, (h_2 + h_1)$$

where h_1 can be calculated in terms of h_2 from Figure 3. (equating the volume displacements)

$$h_1 A_1 = h_2 A_2$$
$$h_1 (\pi/4) d_1^{\,2} = h_2 (\pi/4) d_2^{\,2}$$
$$h_1 = h_2\, (d_2/d_1)^2$$
$$P = k\gamma \{h_2 + h_2\, (d_2/d_1)^2$$
$$= k\gamma\, h_2 \{1 + (d_2/d_1)^2\}$$

The inclined manometer (Figure 4) is quite similar to the well type manometer. The only difference being that its narrow leg is inclined, due to which the pressure is read on a considerably larger scale (depending on the angle). The result of this is a sensitive pressure measuring device capable of detecting smaller pressure changes. Corrections due to changes of level may be necessary unless the diameter ratios of well to tube is sufficiently large. This correction can be evaluated and is similar to that of the well type manometer.

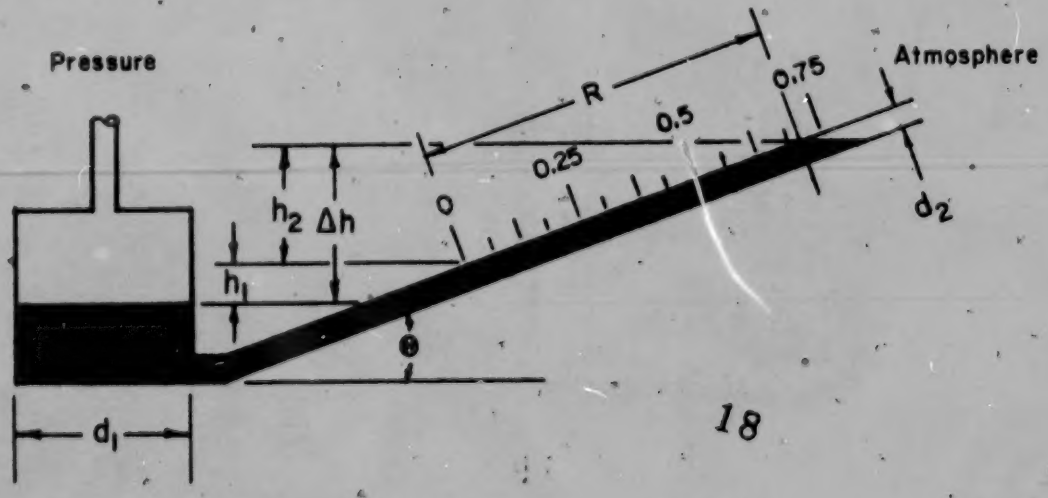

Figure 4. Inclined Manometer

The pressure is calculated by the following equations:

$$P = k \gamma \Delta h$$

$$P = k\gamma \ (h_2 + h_1)$$

$$h_1 d_1^{\,2} = R \, d_2^{\,2}$$

$$h_1 = R \ (d_2/d_1)^2$$

$$h_2 = R \sin \Theta$$

$$P = k\gamma \ \{R \sin \Theta + R(d_2/d_1)^2\}$$

$$P = k\gamma R \ \{\mathrm{Sin} \ \Theta + (d_2/d_1)^2\}$$

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PRESSURE MEASUREMENT EXPERIMENT

OBJECT

To become familiar with the operation and use of various pressure measuring devices.

APPARATUS

3 U-tube manometers
Well type manometer
Inclined manometer
Bourdon tube gage

PROCEDURE

a. Inspect gages noting the level of mercury in the manometers. Take this data as zero reading.

b. Open up discharge valve in manifold wide and start compressor.

c. Close discharge valve slowly, recording data from all instruments at 1 inch increments on well manometer. (Do not exceed 5 psi on test gage).

EVALUATION OF RESULTS

a. Draw a curve of the inclined manometer pressure and the test gage pressure against the well manometer pressure.

b. Compare the devices considering their advantages, accuracies and limitations.

Data Sheet

Well-Type inches Hg	No 1. U-tube inches Hg	No 2. U-tube inches Hg	Test Gage psi	Inclined inches Hg
			0.0	

Specific Gravity of red fluid _____ Height of red fluid _____

Specific Gravity of blue fluid _____ Height of blue fluid _____

Specific Gravity of mercury _____

Inclined manometer angle _____

Diameter ratio well/tube _____

Results Sheet

Well-Type psi	No 1 U-Tube psi	No 2 U-Tube psi	Test Gage psi	Inclined psi
0.0				

The measurement of the area under a curve can be of significant importance in engineering work. The area enclosed by the curve often has physical meaning and needs to be evaluated. An example of this would be the indicator card (Figure 1) which shows the pressure volume variation of the working substance of a reciprocating engine.

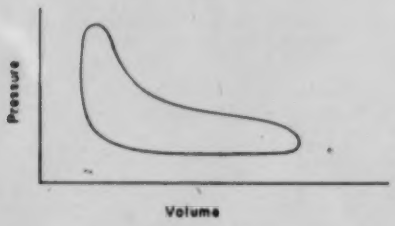

Figure 1. Indicator Card

The evaluation of this area gives the indicated work of the engine. It is also sometimes necessary to evaluate the area under the curve to obtain a truer value of an average property. An example of this would be the calculation of the average velocity in a duct. This is done by plotting the variation of velocity versus position in the duct (velocity profile). The true average velocity can then be calculated by dividing the area under the curve by the position axis.

If the curve is smooth and its equation can be readily determined, then integration ($A = \int y dx$) can be used to evaluate the area exactly. If the curve is irregularly shaped, other means may be employed to find the area. Some of the more common of these methods are the following:

a. Weight Method: The unknown area can be evaluated by setting up a proportionality between the weight of the area under the curve and the weight of a known area cut from the same paper.

b. Average Height: In this method the area under the curve is divided into a number of strips. The average height or mean ordinate of each strip is determined and horizontal lines drawn through them forming several rectangles. The area of each rectangle should approximate the area in the

particular strip and summation of all the rectangles should give the area under the curve. It should be noted that the average height or mean ordinate of each strip depends on the variation of the portion of the curve within the strip (Figure 2).

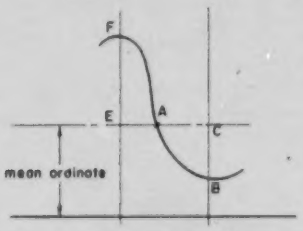

Figure 2. Mean Ordinate (Area ABC = Area AEF)

c. Simpson's Rule: Is a numerical method of integration based on the formula that the area under the arc of a parobola ($y = Ax^2 + Bx + C$) is equal to

$$A_p = h/3 \{y_o + 4y_1 + y_2\}$$

The area under an irregularly shaped curve can be found by Simpson's rule. This is done by dividing the area into an even number of strips an equal distance apart. (Figure 3).

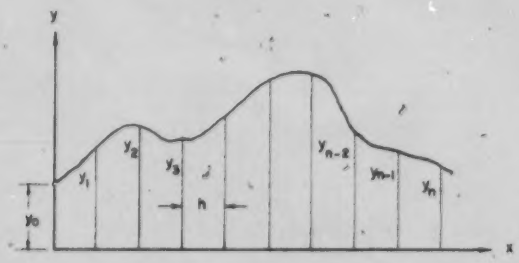

Figure 3. Application of Simpson's Rule

Approximating portions of the curve by parabolas and summing up the results, the following equation can be obtained for the total area under the curve.

$$A = h/3 \{yo + 4y_1 + 2y_2 + 4y_3 + 2y_4 \ldots 2y_{n-2} + 4y_{n-1} + y_n\}$$

d. Planimeter: This is a mechanical integrating device which upon tracing of the boundaries of the curve gives a reading of the enclosed area. The theory and operation of planimeters should be looked up in manufacturers catalogs and other references.

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AREA MEASUREMENT EXPERIMENT

OBJECT

To become familar with four methods of finding the area under an irregularly shaped curve.

APPARATUS

Precision balance
Polar planimeter
Kilowatt versus time curves

PROCEDURE

Using all area measuring methods discussed, determine the area of the energy diagram in square inches and kilowatt-hours. Find the mean ordinate (average height) of the curve in kilowatts for each of the methods used.

Note: Do all work for same energy diagram.

PROCEDURE I

a. Weight Method

1. Cut and weigh on a precision balance the area enclosed by the curve.
2. From same paper, cut and weigh a known area (example: one square inch).
3. Find the unknown area of the curve using a proportion of weights to areas.

b. Average Height

1. Divide the area into an even number of strips of the same width.
2. Find the mean ordinate (average height) of each strip and draw horizontal lines through them forming small rectangles. Refer to Figure 2.
3. Sum up the areas of the rectangles.

c. Simpson's Rule

1. Using the same divisions as in part (b), determine the lengths of the intercepted segments. Refer to Figure 3.

2. Evaluate the area by Simpson's equation.

d. Planimeter

1. Using information in references and manufacturing catalogs, determine area.

PROCEDURE II

a. Convert all the areas evaluated from square inches to kilowatt hours.

b. Determine the mean ordinate, in kilowatts, for each of the methods.

EVALUATION OF RESULTS

a. Tabulate results obtained.

b. Compare methods used, discussing advantages and accuracies of each method.

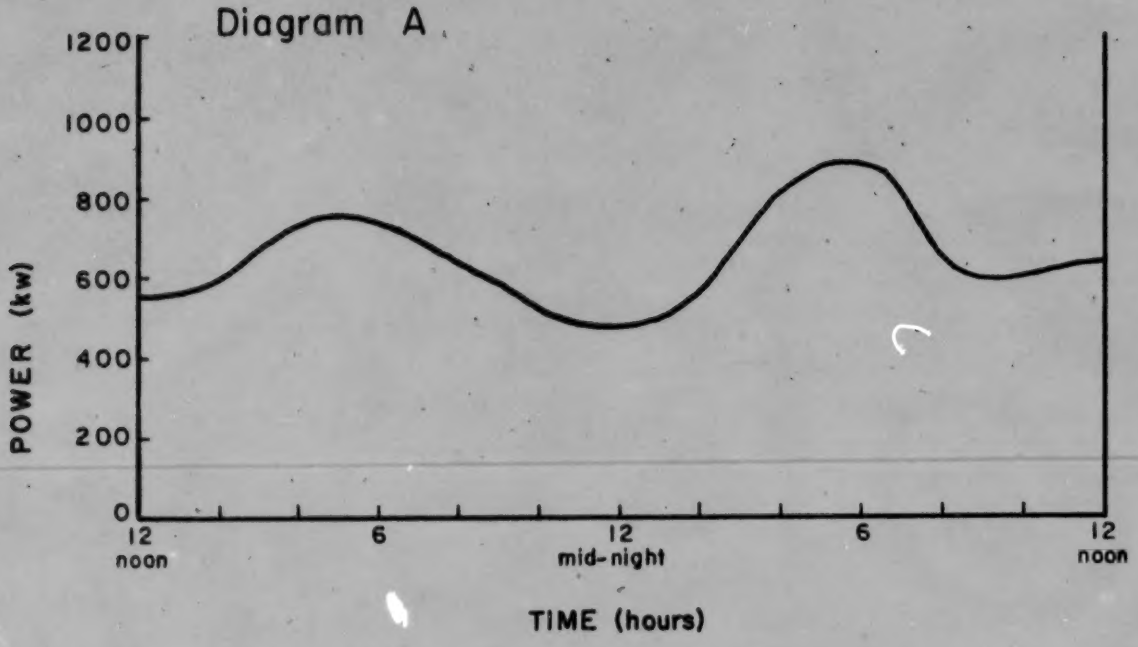


Diagram A
1200
1000
800
600
400
200
0
POWER (kw)
12
noon
6
12
mid-night
6
12
noon
TIME (hours)

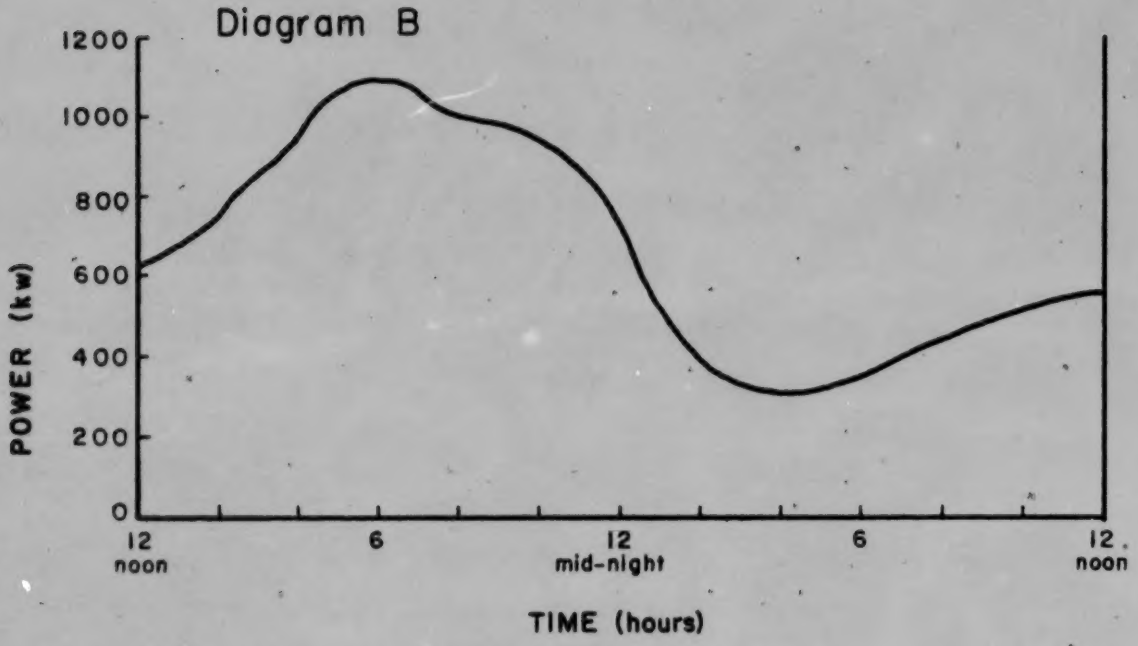


Diagram B
1200
1000
800
600
400
200
0
POWER (kw)
12
noon
6
12
mid-night
6
12
noon
TIME (hours)

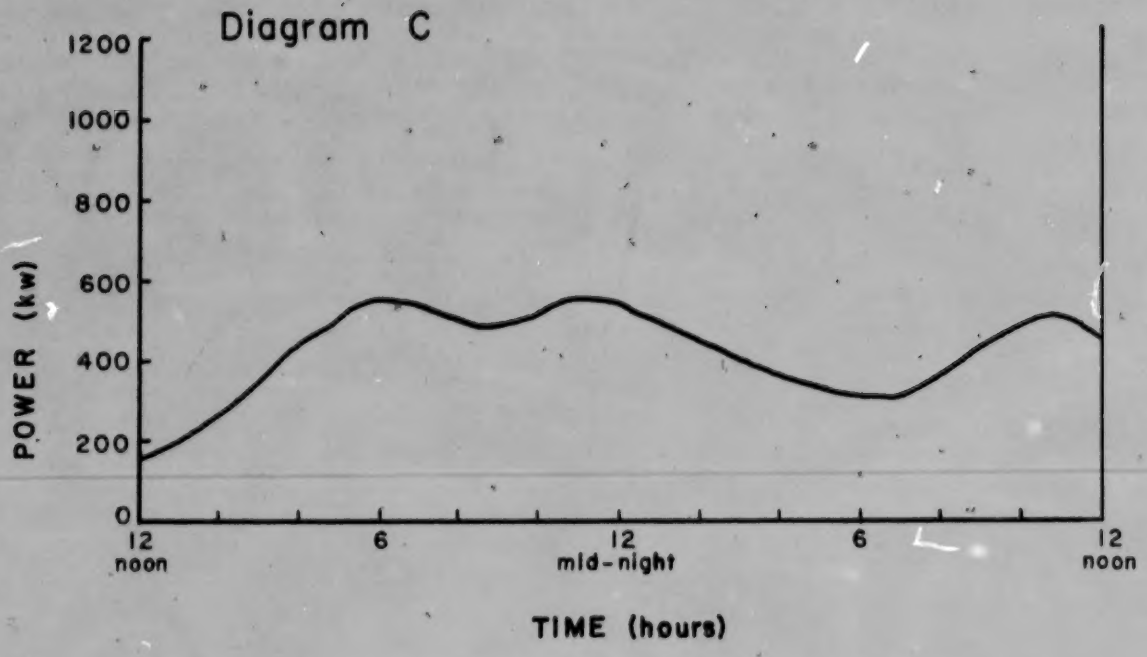


Diagram C
1200
1000
800
600
400
200
0
POWER (kw)
12
noon
6
12
mid-night
6
12
noon
TIME (hours)

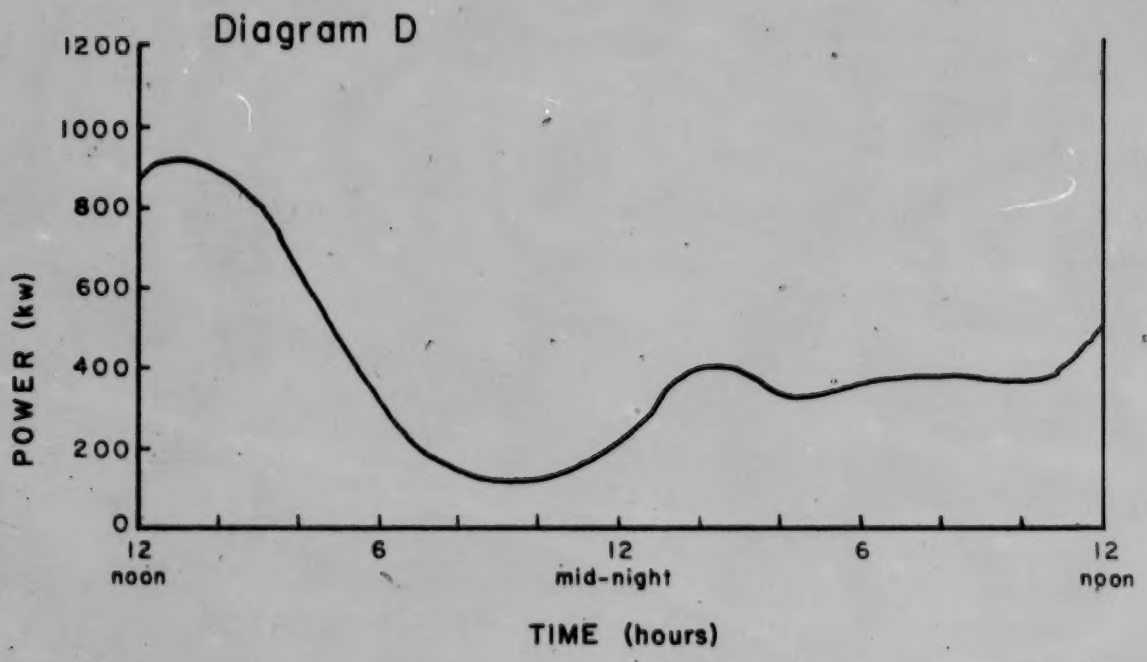


Diagram D
1200
1000
800
600
400
200
0
POWER (kw)
12
noon
6
12
mid-night
6
12
noon
TIME (hours)

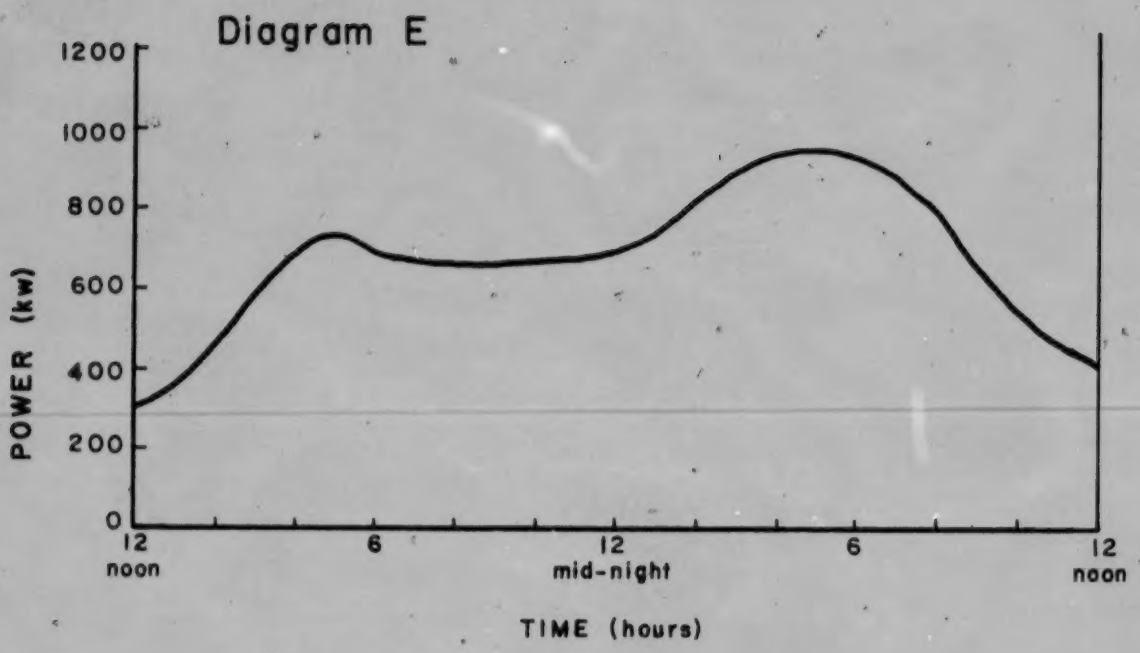


Diagram E
1200
1000
800
600
400
200
0
POWER (kw)
12
noon
6
12
mid-night
6
12
noon
TIME (hours)

Data Sheet

Curve ____

Weight Method

Trial	Weight of Square	Weight of Area
1		
2		

Mean Ordinate Method

Average Height of Strips					
1	2	3	4	5	6
7	8	9	10	11	12

Data Sheet (Continued)

Curve _____

Simpson's Rule Method

h	y_0	y_1	y_2	y_3	y_4	y_5
y_6	y_7	y_8	y_9	y_{10}	y_{11}	y_{12}

h - space dimension

y - ordinate height

Polar Planimeter Method

Trial	Initial Reading	Final Reading
1		
2		
3		

Results Sheet

Method	Area (sq in)	Area (kw hr)	Mean Ordinate (kw)
Weight			
Average Height			
Simpson's Rule			
Polar Planimeter			

The operation of expansion thermometers depend on the volumetric expansion of a liquid in a vacuum (common mercury thermometer), volumetric expansion of a liquid under pressure (gas filled mercury thermometer) and the pressure change of a liquid, gas or vapor in constant volume (filled thermal systems). Glass stem thermometers are graduated for complete immersion unless specified as partial immersion. When using total immersion thermometers in areas where the stem is partially immersed, a correction for stem exposure must be made. This is necessary since all portions of a glass thermometer are sensitive to temperature. When a temperature difference between the bulb and the exposed stem exists, the stem dimensions will change and alter the available liquid space. A second thermometer recommended by ASME test codes and attached, as shown in Figure 1, is used and corrections made by the formula:

$$K = 0.000088D\ (T_1 - T_2)$$

Where D = length of emergent stem in degrees

T_1 = reading of the main thermometer

T_2 = reading of attached thermometer

K = correction in degrees

When the stem is cooler than the bulb, the correction K is added. When warmer, it is subtracted. The bulb of the smaller thermometer should be about three quarters down the exposed mercury column.

The filled thermal systems (pressure gage type) use either a liquid, vapor or gas at constant volume. The system consists of a bulb, capillary tubing and a Bourdon tube measuring element. The temperature sensed by the bulb results in a change of pressure of the fluid in the system. The force exterted due to the pressure change tends to straighten the Bourdon tube and through linkages, this pressure is read as temperature on a calibrated dial.

When two wires of dissimilar metals are connected at both ends a thermocouple is produced. As one junction is heated, an electric current will flow from one to the other across the cold junction. The voltage (emf) produced is almost directly proportional to the temperature differences and can be measured by placing a voltmeter in the circuit. Using appropriate tables the temperature can be determined for a given voltage reading.

The potentiometer is the most common instrument used to measure the emf or thermocouple output produced. This instrument consists mainly of a calibrated slide wire resistance, a galvanometer and a battery which acts against the current flow from the thermocouple. When the thermocouple emf is balanced against this known voltage, there is no current flow in the thermocouple circuit and any resistance in the leads is largely eliminated. In addition to the above, a standard cell which maintains its voltage for several years is included for calibration along with either a manual or automatic cold junction compensator for use when the unlike wires of the thermocouple are connected to the binding posts of the instrument.

A resistance thermometer, although not used in this experiment, has as its sensing element a resistor. These resistors are generally made with a good conducting pure metal in the form of wire (platinum, nickel, copper) and the resistance produced is directly proportional to the temperature change. Thermistors which are made from ceramic bead mixtures of metallic compounds have negative resistance coefficients of a large magnitude, and when used with the correct indicating instruments are highly sensitive and accurate. Other types of instruments would be the bimetallic dial thermometer, the optical pyrometer and the radiation pyrometer.

REFERENCES

Ambrosius, E. E., Fellows, R. D. and Brickman, A. D., "Mechanical Measurement and Instrumentation", Ronald Press, Inc., New York, 1966.

Beckwith, T. G. and Buck, N. Lewis, "Mechanical Measurements", Addison-Wesley, Inc., Reading, Massachusetts, 1961.

Shoop, C. F. and Tuve, G. L., "Mechanical Engineering Practice", McGraw Hill, New York, 1956, Fifth Edition.

TEMPERATURE MEASUREMENT EXPERIMENT

OBJECT

a. To compare the accuracies of various temperature measuring devices with a commercial standard.

b. Calibrate a vapor pressure thermometer.

APPARATUS

2 - 0° - 220° total immersion thermometers
1 - Vapor filled thermometer
1 - Portable Potentiometer
2 - Copper - Constantan thermocouples
1 - Safety beaker 3000 ml capacity

PROCEDURE

a. Place the pyrex beaker filled with water on an electric heating plate.

b. Immerse the two total immersion type thermometers as shown in Figure 1. Immersion point of T_1 should be at 40 degrees.

c. Using the thermocouples furnished, connect the leads to the potentiometer as shown in Figure 2.

d. Avoiding contact with the walls of the beaker, attach the thermocouple end at the measuring junction on thermometer bulb.

e. Place the bulb of the vapor thermometer into the water next to the mercury thermometer.

f. Turn on heating element and slowly heat the water taking readings of all instruments at 10° intervals on T_1.

g. Heater setting should allow 10° increase in 3 to 4 minutes.

h. References should be made to the operation manual for all instruments used.

EVALUATION OF RESULTS

Using the corrected thermometer reading as the standard, compare and analyze all readings together with any information obtained from the following graphs.

a. Plot a curve with values taken from the millivolt conversion tables of millivolts as abscissa versus temperature as ordinate.

b. Plot a zero error curve of temperature. Use same scale of temperature as abscissa and temperature as ordinate. (Curve should be a 45° line). On the same coordinate system, plot (points only) vapor thermometer reading as abscissa versus corrected thermometer reading as ordinate.

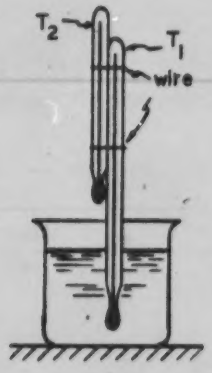
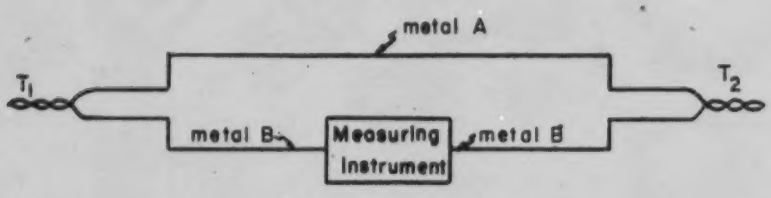


Figure 1. Arrangement of Immersion and Auxiliary Thermometers.



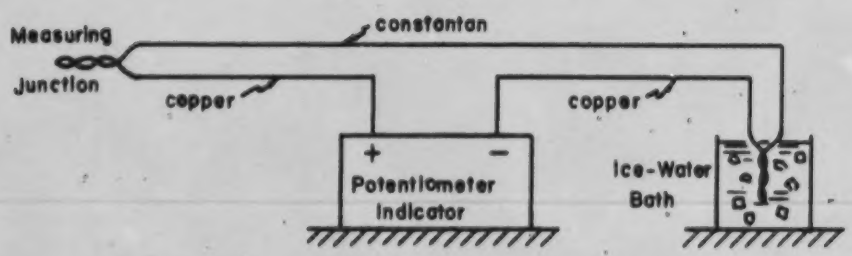


Figure 2. Connections for Thermocouples Using Ice Water Reference Junction.

Data Sheet

Temp. T_1 (°F)	Temp. T_2 (°F)	Meter Reading (mv)	Gage Reading (°F)
80			
90			
100			
110			
120			
130			
140			
150			
160			
170			
180			
190			
200			
210			
212			

Point of Immersion of T_1 _____

Barometric Pressure _____

Results Sheets

Temp. T_i (°F)	Correction K	Corrected Temp. (°F)	Conv. Ind. Reading (°F)	Gage Reading (°F)
80				
90				
100				
110				
120				
130				
140				
150				
160				
170				
180				
190				
200				
210				
212				

SPECIFIC GRAVITY MEASUREMENT — EXPERIMENT NO. 5

Specific gravity is the ratio of the density or specific weight of a fluid to the density or specific weight of water at $60^{\circ}F$.

The density ρ(rho) of a fluid is its mass per unit volume, whereas the specific weight γ (gamma) is expressed in pounds force per unit volume.

$$\text{Density } (\rho) = \text{lbm/ft}^3$$

$$\text{Specific weight } (\gamma) = \text{lbf/ft}^3$$

Referring to our definition of specific gravity, we find that by knowing the density or specific weight of water at the reference temperature of $60^{\circ}F$. we can readily find the density of any other fluid by knowing its specific gravity.

Specific gravity is usually determined through the use of a hydrometer or hydrostatic balance. A hydrometer (Figure 1) is a calibrated float which has a scale indicating specific gravity directly or in degrees.

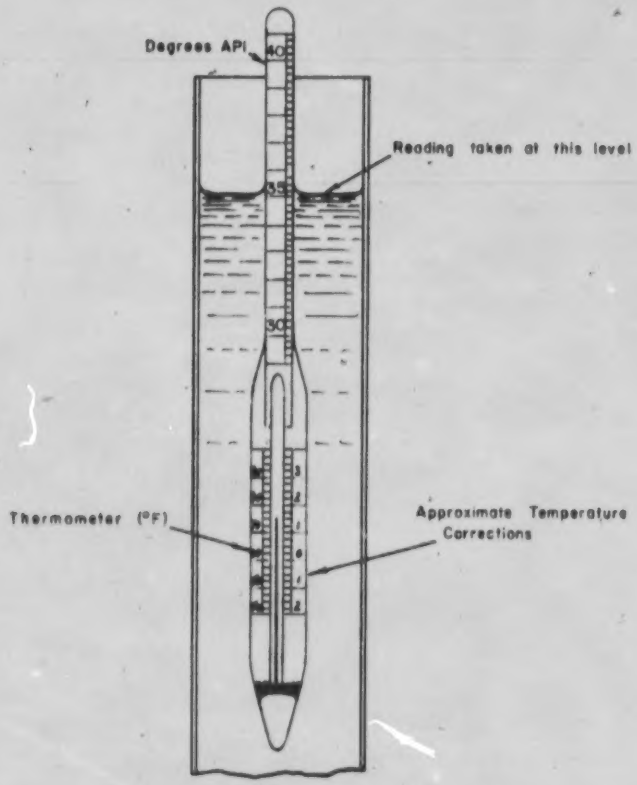


Figure 1. API Hydrometer

A reading is obtained on the scale at the level of the liquid. For better accuracy, the two scales which are widely used on hydrometers are the Bureau of Standards or Baume and the A.P.I. (American Petroleum Institute). The relationship between degrees and specific gravity for the two scales is

$$\text{Specific gravity } 60/60^{\circ}\text{F} = \frac{140}{130 + {}^{\circ}\text{Be @ } 60^{\circ}\text{F}}$$

$$\text{Specific gravity } 60/60^{\circ}\text{F} = \frac{141.5}{131.5 + {}^{\circ}\text{API @ } 60^{\circ}\text{F}}.$$

The ratio of the density of the fluid at 60°F. to the density of water at 60° is shown by the term $60/60^{\circ}$F. If the liquid cannot be brought to 60°, correction must be made with the use of suitable tables. All hydrometers are usually calibrated to give readings corrected for the buoyant effect of air.

The hydrostatic balance depends upon the determination of weights required to counteract the buoyant effect of a liquid upon a totally immersed plummet of fixed volume. The theory and operation of hydrostatic balances should be looked up in manufacturers cataloges and other references.

REFERENCES

Albertson, Barton, Simons, "Fluid Mechanics For Engineers", Prentice Hall, Inc., New Jersey, 1966.

ASTM - IP, "Petroleum Measurement Tables", American Edition, United States Units of Measurement, Philadelphia, 1952.

Binder, R. C., "Fluid Mechanics", Prentice Hall, Inc., New Jersey, 1962, Fourth Edition.

Tuve, G. L., "Mechanical Engineering Experimentation", McGraw Hill, Inc., New York, 1966.

Vennard, J. K., "Elementary Fluid Mechanics", John Wiley & Sons, Inc., New York, 1966, Fourth Edition.

SPECIFIC GRAVITY EXPERIMENT

OBJECT

To determine the specific gravity of different fluids by various accepted methods.

APPARATUS

Hydrometer Cylinder
A.P.I. Hydrometers
Direct reading hydrometer
Thermometers
Hydrostatic Balance

GENERAL INFORMATION

a. Pour sample to be tested into the cylinder without splashing. This will help to eliminate the formation of bubbles. If bubbles do form, they should be removed before proceeding.

b. The hydrometer should be lowered to a level two smallest scale divisions above that at which it will float and then released.

c. When hydrometer has come to rest and is floating freely, read the gravity at the point at which the surface of sample apparently cuts into the hydrometer scale, as shown in Figure 1.

PROCEDURE

a. Hydrometer Method

1. Pour sample into cylinders observing precautions as outlined in preceeding discussion.

2. With hydrometers and thermometers provided, determine the A.P.I. gravity. The specific gravity and the temperature of each sample to be tested.

b. Hydrostatic Balance Method

1. Level the balance and suspend the glass plummet in air from the end of the beam. (Pointer connected to beam should be at its center point).
2. Totally immerse the plummet into the sample.
3. Move weight from zero position on balance arm until plummet just rises.
4. Adjust the vernier scale until plummet is fully balanced.
5. The specific gravity is now determined by adding the vernier reading to the number at the weight position on the arm.
6. Repeat above procedure for all fluids listed on data sheet.

EVALUATION OF RESULTS

a. Calculate the specific gravity for each sample tested using the A.P.I. gravity corrected to $60^{\circ}F$.

b. Compute the density in pounds mass and slugs per cubic foot for each sample taken from corrected hydrostatic balance reading.

c. Discuss any errors or discrepancies that exist between experimental values and data from engineering handbooks.

Data Sheet

Fluid		Specific Gravity		Degrees API	Temperature Degrees F
		Hydrometer	Balance		
OIL	SAE 10				
	Linseed				
	SAE 30				
Diesel Fuel					
Kerosene					
Tap Water					

Results Sheets

Fluid		API	Specific Gravities			Density lb/cu.ft.	Density slugs/cu.ft.
			From API	Hydrometer	Balance		
Oil	SAE 10						
	Linseed						
	SAE 30						
Diesel Fuel							
Kerosene							
Tap Water							

All values corrected to 60°F.

SPEED MEASUREMENT

EXPERIMENT NO. 6

Speed can be defined as the time rate of motion. This motion may be linear, angular or some combination of the two. Four general types of instruments for obtaining speeds of rotating devices that are commercially available, are described as follows:

1. Mechanical Counters and Timers: The revolution counter which is either a hand or clutch connected counter is generally manually timed with a stop watch from which starting and stopping errors may result. The tachoscope in which time and motion are automatically integrated, consists of a stop watch built in with a revolution counter. A push button starts the watch and counter elements simultaneously. This eliminates human error and is claimed to have a high degree of accuracy.

2. Tachometers: The centrifugal weight of common hand tachometer depends upon the centrifugal force moving a weight against the action of a spring. The position of the weight is a function of the centrifugal force and hence the speed. Most tachometers of this type are sensitive, respond to variations in speed and are available in several speed ranges. The advantage of the tachometer over the counter-stop watch method is that it will indicate whether or not the speed remains constant. By substituting a suitable rubber tired wheel of known diameter or circumference for the conical tip, revolutions per minute, peripheral and linear speeds of belts and surface speeds of pulleys may be found using the following data and equations.

Let a = Outside diameter of rubber wheel.
b = Circumference of rubber wheel.
r = RPM shown by tachometer.
D = Diameter of cylinder in inches.
C = Circumference of cylinder in inches.
R = Flywheel RPM

To determine R.P.M. (R).

$$R = \frac{rxa}{D} = \frac{rxb}{C}$$

To determine peripheral or linear speeds in feet per minute(S).

$$S = \frac{RxC}{12}$$

3. Strobotac: The strobotac is a flashing light source utilizing the persistance of vision when an object is received intermittently. It produces an optical effect of stopping a mark on a rotating shaft at the same point in each cycle. By changing its frequency slightly, slow motion can be obtained. Each strobotac should be calibrated to its voltage supply before being put into use. Speeds between 600 and upwards to approximately 40,000 RPM are rapidly obtained with this type of tachometer.

4. Electric Counters and Tachometers: Pulse counters and voltage generators are the two general types. Electronic pulse counters use a pick-up that generates a wave or pulse for each event to be counted. A converter which changes frequency to voltage allows the unit to indicate revolutions per minute continuously. Direct or alternating current voltage generators have voltmeters calibrated directly in RPM.

Other instruments of importance, but not mentioned above, are the vibrating reed tachometer, the chronotachometer and the photo electric tachometer.

In selecting the proper type of instrument, consideration must be given to factors involving cost, portability if needed, accuracy desired, speed to be measured and the size of the rotating element.

REFERENCES

Ambrosius, E. E., Fellows, R. D. and Brickman, A. D., "Mechanical Measurement and Instrumentation", Ronald Press, Inc., New York, 1966.

Shoop, C. F. and Tuve, G. L., "Mechanical Engineering Practice", McGraw Hill, New York, 1956, Fifth Edition.

Tuve, G. L., "Mechanical Engineering Experimentation", McGraw Hill, New York, 1961.

SPEED MEASUREMENT EXPERIMENT

OBJECT

To become familiar with the operation and use of various speed measuring devices and to compare their accuracy against a standard.

APPARATUS

Hand tachometer with accessories
Revolution counter
Stop watch
Strobotac

PROCEDURE

a. Calibrate strobotac using instructions set forth in the instruction manual.

b. Using the strobotac, set the rheostat which controls the motor speed to maximum rpm. Take readings using the revolution counter and stop watch, the hand tachometer and the hand tachometer with rubber wheel.

c. Decrease in increments of 200 rpm and repeat for eight other settings of the rheostat.

d. Calculate the shaft rpm using data from instruments which do not read rpm directly.

e. Calculate the peripheral speed of the motor wheel in ft. per minute using the strobotac data.

EVALUATION OF RESULTS

a. Draw a zero error curve of rpm versus rpm using the same scale for ordinate and abscissa. (Curve should be 45° line).

b. On the same set of coordinates, plot strobotac rpm as abscissa versus revolution counter and tachometer rpm as the ordinate.

c. Analyze the results and explain any errors or discrepancies as each instrument is compared to the strobotac.

SPEED MEASUREMENT EXPERIMENT

OBJECT

To become familiar with the operation and use of various speed measuring devices and to compare their accuracy against a standard.

APPARATUS

Hand tachometer with accessories
Revolution counter
Stop watch
Strobotac

PROCEDURE

a. Calibrate strobotac using instructions set forth in the instruction manual.

b. Using the strobotac, set the rheostat which controls the motor speed to maximum rpm. Take readings using the revolution counter and stop watch, the hand tachometer and the hand tachometer with rubber wheel.

c. Decrease in increments of 200 rpm and repeat for eight other settings of the rheostat.

d. Calculate the shaft rpm using data from instruments which do not read rpm directly.

e. Calculate the peripheral speed of the motor wheel in ft. per minute using the strobotac data.

EVALUATION OF RESULTS

a. Draw a zero error curve of rpm versus rpm using the same scale for ordinate and abscissa. (Curve should be 45° line).

b. On the same set of coordinates, plot strobotac rpm as abscissa versus revolution counter and tachometer rpm as the ordinate.

c. Analyze the results and explain any errors or discrepancies as each instrument is compared to the strobotac.

Data Sheet

Strobatac rpm	Tachometer rpm	Tach with Rubber Wheel	Revolution Counter	Time Seconds

Diameter of Motor Flywheel _____

Diameter of Rubber Wheel _____

Results Sheet

Flywheel rpm				Peripheral Speed of Flywheel ft/min
Using Strobofac	Using Tachometer	Using Rubber Wheel	Using Rev. Counter	

PRESSURE - TEMPERATURE RELATIONSHIP

EXPERIMENT NO. 7

There is a precise temperature at which a liquid will begin to boil dependent on its pressure. This boiling point temperature is referred to as the saturation temperature and the pressure is designated as its saturation pressure. It is possible to develop on an experimental basis a table of pressure-temperature relationships by carefully heating a fluid at some predetermined pressure and recording the temperature at which the liquid begins to boil. The reverse of this is also true. If a saturated liquid is allowed to cool and pressure above is reduced, eventually the liquid will boil at some lower temperature.

A fluid with a saturation temperature lower than ambient (a refrigerant) will boil at its saturation temperature if exposed to the atmosphere. This liquid when placed in a constant volume container and allowed to reach thermal equilibrium with its surroundings will develop a pressure equal to its saturation pressure. Every fluid has this unique characteristic and because of this relationship it is possible to identify the particular fluid from these two properties.

REFERENCES

Air Conditioning and Refrigeration Institute, "Properties of Commonly-used Refrigerants", Air Conditioning and Refrigeration Institute, Arlington, Virginia, 1967.

Keenan, J. H., Keys, F. G., Hill, P. G., Moore, J. G., "Steam Tables", John Wiley and Sons, Inc., New York, New York, 1969.

Van Wylen, G. J., "Thermodynamics", John Wiley and Sons, Inc., New York, New York, 1959.

PRESSURE-TEMPERATURE RELATIONSHIP EXPERIMENT

(PART A)

OBJECT

To investigate the effect of pressure change on the boiling point of liquids.

APPARATUS

2000 ML - Erlenmeyer Flask
Vacuum-pump
Electric heater
Thermometer
Vacuum gage

PROCEDURE

a. Set up equipment as indicated in Figure 1.

b. Preheat 1800 milliliters of fluid to 100°F. with maximum heat setting on electric heater.

c. Open control valve "X" to atmosphere and start vacuum pump. Close control valve and allow pump to pull maximum vacuum (27").

d. With pressure maintained constant, add heat slowly (set rheostat to give 3°F. - 4°F. per minute rise) until water begins to boil* and at this point record the temperature.

e. Crack open valve "X" until the pressure is increased by 3" Hg. (Note what happens to the boiling water).

f. Repeat step d and continue method as outlined in step d and step e until pressure in vessel is atmospheric.

* This point is critical if reliable data is to be obtained. (Consider boiling temperature to be at that point when bubbles form at surface of liquid and close observation indicates that vapor is beginning to be formed above the liquid).

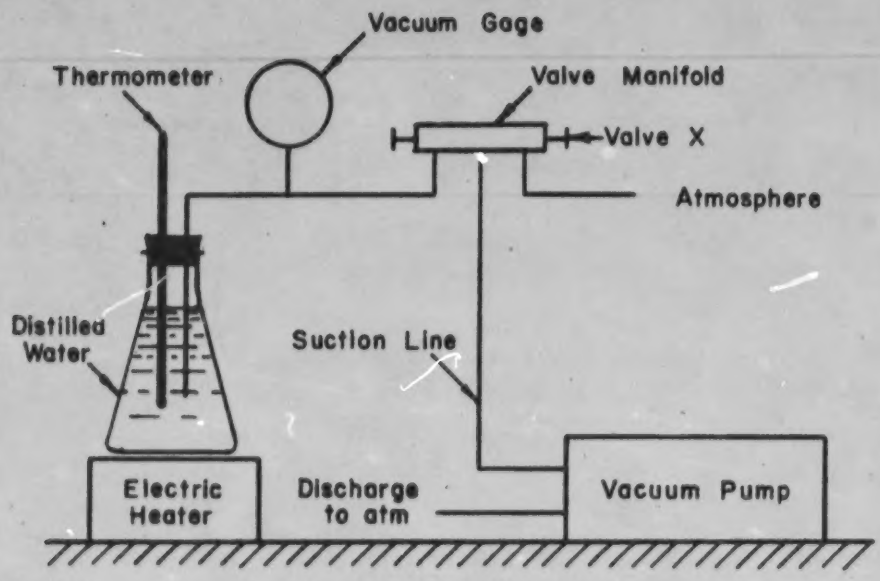


Figure I. Boiling Point Apparatus

EVALUATION OF RESULTS

a. Using appropriate tables, plot a curve of temperature as ordinate versus absolute pressure as abscissa for each fluid tested. These curves will now be used as a standard around which experimental points may be plotted.

b. Discuss the reasons for any discrepancies that may exist between experimental points and tabular values.

PRESSURE-TEMPERATURE RELATIONSHIP EXPERIMENT

(PART B)

OBJECT

To investigate the effect of pressure changes on the boiling point of liquids.

To identify various refrigerants from their pressure temperature relationship.

APPARATUS

2 cans of refrigerant
Temperature Gage
Pressure Gages

PROCEDURE

a. Open valve on each can of refrigerant and check systems for leaks with a soap-suds solution.

b. Adjust hot and cold water streams so that initial temperature of bath water is at $120^{\circ}F$. (under no conditions should temperature of bath water exceed $125^{\circ}F$.).

c. Once water is at $120^{\circ}F$, allow the refrigerant in the containers and the surrounding water to reach equilibrium. (This can be obtained by using some form of paddle to agitate the water).

d. When it is felt that equilibrium conditions have been reached, record the pressure for each refrigerant at this temperature.

e. Reduce the temperature of the bath water in increments of $5^{\circ}F$. and at each of these intervals record the corresponding pressures.

f. Repeat this procedure until system reaches $70^{\circ}F$.

Data and Results Sheet

Part "A"

Inches Hg Vácuum	Absolute Pressure	Saturation Temperature	
		Experimental	Tables

Data and Results Sheet

Part "B"

Bath Water Temperature degrees F	Refrigerant Pressure (psig)			
	X		Y	
	Experimental	Tabular	Experimental	Tabular

CALIBRATION OF INCLINED MANOMETER

EXPERIMENT NO. 8

Many of the instruments used to measure velocity are actually pressure sensing devices. In most cases, differential pressure is necessary for either direct reading of velocity or conversion to obtain the unknown velocity (i.e.; pitot-tube). To insure accuracy, these gages must be periodically calibrated.

The Bourdon Tube, U-tube manometer, inclined manometer and Hook gage micro-manometer are used extensively as differential pressure measuring devices. With the exception of the Hook gage, these instruments have been discussed in previous experiments.

The Hook gage is an instrument designed to read minute changes in air pressure with extreme accuracy. The scales read from 0 to 2 inches or 0 to 4 inches of water (usually) in hundredths of an inch and in thousandths of an inch on a vernier scale. They are highly suited for test work in industry and provide a standard for the calibration of other instruments.

The most common type of Hook gage is constructed with two glass vials containing distilled water. A hook or needle point is moved by a micrometer adjustment until the point pierces the water surface but does not break the surface tension. The difference in the micrometer readings gives the differential pressure.

CALIBRATION OF INCLINED MANOMETER EXPERIMENT

OBJECT

To calibrate pressure differential measuring devices.

APPARATUS

Dwyer Hook Gauge

2 - well type inclined manometers

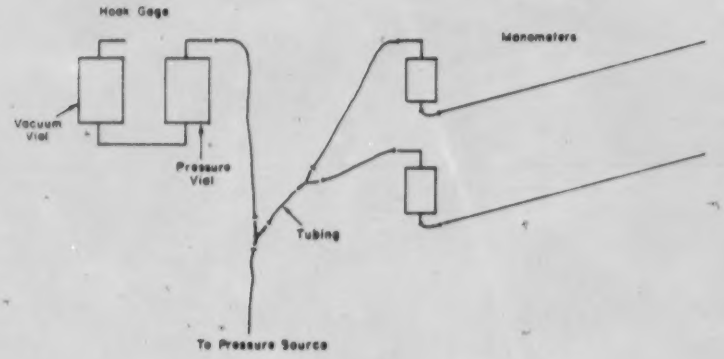
Fan test set-up

PROCEDURE

a. Adjust Hook gauge for zero reading.

b. Determine zero reading for inclined manometer.

c. Attach connections as shown in accompanying diagram.



d. Set fan at maximum speed with duct exit wide open.

e. Adjust Hook gauge for new reading. Record values for both pressure and vacuum readings along with readings for manometer #1 and manometer #2.

f. Increase pressure by decreasing duct exit opening with throttling devices provided. Record values on data sheet.

g. Repeat step "f" for all openings.

EVALUATION OF RESULTS

a. Determine angle of inclination for both manometers.

b. Plot a graph of Hook gauge readings as ordinate versus manometer readings as abscissa.

c. What would be the advantages of this method of calibration?

Data and Results Sheet

Hook Gage			Manometers	
Pressure	Vacuum	Difference	No. 1	No. 2

Barometric Pressure _____ Air Temperature _____

	Manometer No. 1	Manometer No. 2
Initial Readings	_____	_____
Sp. Gr. of Liquids	_____	_____
Inclination Angle	_____	_____

There are various devices, commercially available that are used extensively to measure the velocity of a fluid. The selection of the proper device depends on factors, such as, type of flow, closed conduit or open channel, type of fluid, velocity range, accuracy and cost.

Pitot tubes are probably the most commonly used instruments for measuring velocity. They are pressure sensing devices that measure point velocities indirectly by measuring the velocity pressure at that point. To discuss the theory of pitot tubes, it is essential to understand the terminology associated with the measurement of pressure.

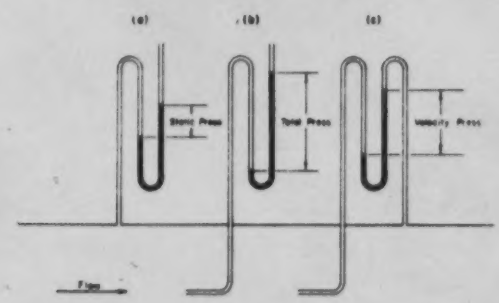


Figure 1. Static, Total, and Velocity Pressures

It can be seen from Figure 1 (c) that the velocity pressure is the difference between total and static pressure.

The simple pitot tube is a tube bent at right angles and tapered at one end as shown in Figure 2.

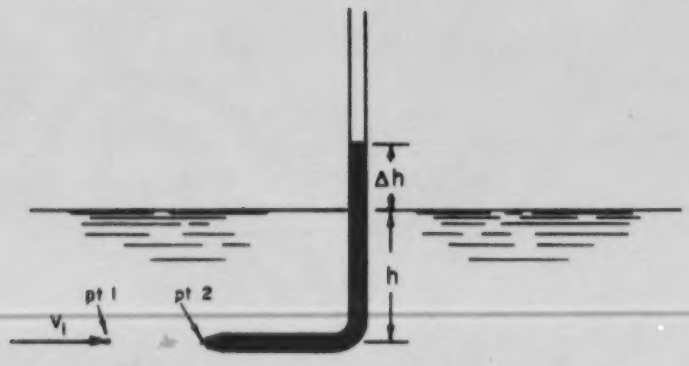


Figure 2. Simple Pitot Tube

The tapered end is pointed upstream parallel to the flow and the instrument at point 2 measures the total or stagnation pressure $(h + \Delta h)$. It can be seen from the figure that the static pressure contribution is equal to (h) while the velocity pressure contribution is equal to (Δh), since:

$$P_{(total)} = P_{(static)} + P_{(velocity)}$$

$$P_{(total)} = h + \Delta h$$

The velocity at point 1 can now be calculated by applying the Bernoulli equation along the streamline passing through points 1 and 2.

$$P_1/\gamma + V_1^2/2g + Z_1 = P_2/\gamma + V_2^2/2g + Z_2 \qquad (1)$$

where P = pressure in lbf/ft^2

γ = specific weight in lbf/ft^3

V = velocity in ft/sec

g = local acceleration of gravity in ft/sec^2 (standard value of g = 32.2 ft/sec^2)

Z = elevation in ft.

Dimensionally using specific units each term in the Bernoulli equation has units of elevation in feet, and represents a form of energy. The total energy (summing up all the terms) stays constant along the streamline.

Since $Z_1 = Z_2$ (same elevation) and the velocity at point 2 is zero (Stagnation point), equation 1 may be simplified as follows:

$$P_1/\gamma + V_1^2/2g = P_2/\gamma \qquad (2)$$

Defining the static pressure head $P_1/\gamma = h$, and the total pressure head $P_2/\gamma = h + \Delta h$, permits a rewriting of equation 2 in the following form:

$$h + V_1^2/2g = h + \Delta h$$

Therefore:

$$V_1^2/2g = \Delta h$$

$$V_1 = \sqrt{2g\Delta h}$$

The simple pitot tube previously discussed is a total pressure measuring device and certain difficulties are encountered in measuring the velocity pressure (Δh). The pitot-static tube or combined pitot tube makes the measurement of velocity somewhat easier. This instrument is actually a combination of a static tube which measures the pressure of the undisturbed fluid (static pressure) and a simple pitot tube as shown in Figure 3.

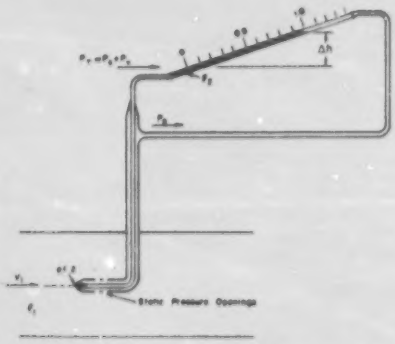


Figure 3 Pitot-Static Tube

The pitot static tube is capable of measuring static pressure, total pressure, or the difference of the two, velocity pressure. The velocity can again be calculated by applying the Bernoulli equation and the result is the same as that of the simple pitot tube.

$$V_1 = \sqrt{2g\ (\Delta h)}$$

Δh in this relationship is in height of indicator fluid (in manometer) and an equivalent height in feet of fluid in the system can be evaluated by multiplying Δh by the density ratio of the fluids. For example:

if air is flowing in the duct and water is the indicating fluid in the manometer

$$V_1 = \sqrt{2g\ \Delta h\ \frac{\rho 2}{\rho 1}} \qquad (3)$$

ρ_2 = density of water

ρ_1 = density of air

It should be noted that the inclined manometer scale can be calibrated in velocity units and give direct readings of velocity.

Pitot Cylinders and Spheres - It is often necessary, in addition to the magnitude of velocity, to find the direction of velocity. Pitot cylinders for two dimensional flow and pitot spheres for three dimensional flow (Figure 4) are especially suited to find direction.

$P_A = P_C$ for Cylinder

$P_A = P_B$, $P_C = P_D$ } for Sphere

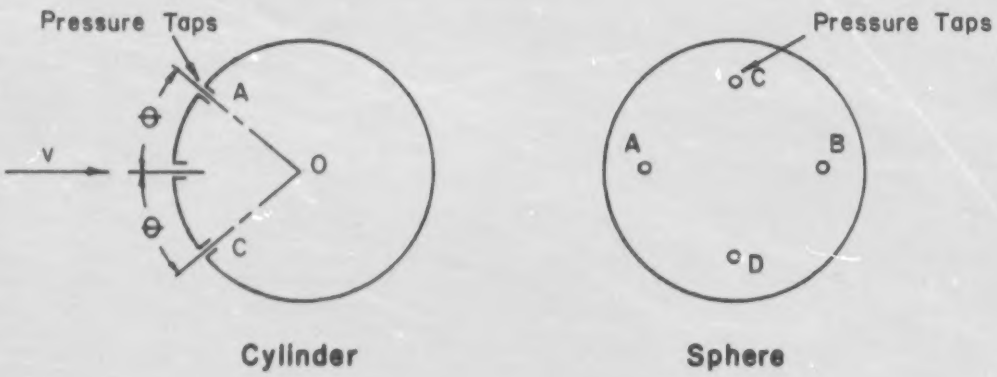


Figure 4. Pitot Cylinder and Pitot Sphere

The instruments basically work on the same principle. A study of Figure 4 shows that the direction of velocity can be determined by adjusting the position of the cylinder and until the pressure difference is zero.

Mechanical Anemometer - is made of a number of blades or cups which are mounted on a shaft (like a fan) and are free to rotate. The velocity is measured by the speed of rotation of the blades. The blades and cup are designed such that they offer little resistance to flow (precision bearings, low inertia) and with calibration they can determine wind or duct velocities.

Alnor Velometer - is an air velocity measuring device. It is based on the pitot tube principle where the velocity pressure is somehow used to give a direct measure of velocity on a calibrated scale. It is primarily used to measure velocities in heating and air conditioning fields.

Dwyer Air Meter - is a direct air velocity measuring device. It is based on the variable area flowmeter principle (rotameter) and is calibrated to give direct readings of velocity.

REFERENCES

Fribance, A. E., "Industrial Instrumentation Fundamentals", McGraw-Hill, New York, 1962.

Streeter, V. L., "Fluid Mechanics", McGraw-Hill, New York, Fifth Edition, 1971.

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VELOCITY MEASUREMENT EXPERIMENT

OBJECT

To become familiar with various air velocity measuring devices.

APPARATUS

Centrifugal fan and duct system
Pitot-static tube and differential manometer
Mechanical anemometer
Alnor velometer
Dwyer air-meter
Strobotac

PROCEDURE

a. Set-up pitot-tube and differential manometer as shown in Figure 3.

b. Operate fan at maximum speed. Using the various velocity measuring devices, take and record readings at center of duct.

c. Reduce fan speed in increments of 100 RPM and take velocity readings at each interval.

d. Calculate the velocity from pitot-static tube data.

EVALUATION OF RESULTS

a. Plot a zero error curve of velocity. Use same scale for velocity as abscissa and velocity as ordinate. (Curve should be a 45° line). On the same coordinates plot (points only):

   1. Mechanical anemometer readings as abscissa versus pitot tube as ordinate.

   2. Dwyer gage readings as abscissa versus pitot tube as ordinate.

   3. Velometer reading as abscissa versus pitot tube as ordinate.

b. Discuss the advantages and disadvantages of each device considering such factors as cost, accuracy, etc.

Data and Results Sheet

Fan RPM	Pitot-Tube			Mechanical Anemometer fpm			Dwyer Air Meter fpm	Alnor Velometer fpm
	Man. Read.	in. water	fpm	data	correction	Results		

Air Temperature _____ Barometric Pressure _____

Specific Gravity of Manometer Fluid _____

There are two different types of flow: laminar and turbulent. In laminar or viscous flow the fluid moves in layers; each layer moving smoothly alongside its adjacent layer in an approximate parallel path. The particles contained in a given layer do not mix with those of an adjacent layer. In turbulent flow the motion becomes irregular, the particles start moving at random in different directions, resulting in their mixing. The fluid, in order to proceed from laminar to turbulent flow regime, has to go through a transition zone. In the transition zone the fluid could be either laminar or turbulent or could intermittently change between laminar and turbulent.

To study flow of fluids it is necessary to determine whether the flow is laminar or turbulent. In the case of closed conduits, such as pipes or ducts, the pressure drop due to frictional losses is a function of the type of flow and would be significantly different for either regime.

Osborne Reynolds, in his study of closed conduit flow, developed a dimensionless parameter which allows one to apply limited experimental results to a wide number of similar situations. The results of his work showed that the following variables, density (ρ), velocity (V), diameter of pipe (D), and viscosity of the liquid (μ) could play significant roles in changing the flow from laminar to turbulent. A change in any one of these variables could effect the type of flow.

$$Re = \frac{\rho V D}{\mu}$$

Reynolds Number can now be used to determine whether the flow in a piping system is laminar or turbulent. Experimental results give the following values as guide lines for usual conditions of pipe flow.

$Re < 2000$ Laminar flow
$2000 < Re < 4000$ Transitional flow
$Re > 4000$ Turbulent flow

A more generalized study of the Reynolds Number reveals that it could be expressed as a ratio of the following forces in a fluid stream

$$Re = \frac{\text{Inertia Forces}}{\text{Viscous Forces}}$$

where the inertia forces are the forces necessary to overcome the resistance of the fluid to motion

$$F(\text{inertia}) = \dot{m}V = (\rho\, AV)V$$

and the viscous forces are the forces necessary to overcome the internal frictional effects or shearing resistances for the fluid area. These forces depend on the viscosity of the fluid

$$F(\text{viscous}) = \frac{\mu AV}{L}$$

$$Re = \frac{(\rho AV)V}{\mu AV/L} = \frac{\rho VL}{\mu}$$

where L is a characteristic length (for pipes L = D).
By using the following dimensions for the above variables a dimensionless number can be obtained.

$$Re = \frac{(\text{slug/ft}^3)\ (\text{ft/sec})\text{ft}}{\text{Slug/ft-sec}}$$

The equation could be dimensionally simplified if kinematic viscosity ν(nu) is used instead of absolute viscosity. Since kinematic viscosity equals μ/ρ, Reynolds Number equation becomes

$$Re = \frac{VL}{\nu} = \frac{(\text{ft/sec})\ (\text{ft})}{\text{ft}^2/\text{sec}}$$

It can be seen that for a small Reynolds Number the prevailing forces are viscous, while for a large number inertia forces prevail.

REFERENCES

Binder, R. C., "Fluid Mechanics", Prentice-Hall, Inc., New Jersey, 4th Edition.

Streeter, V. L., "Fluid Mechanics", McGraw-Hill, Inc., New York, 1971, 5th Edition.

Swanson, W. M., "Fluid Mechanics", Holt, Rinehart and Winston, New York, 1970.

REYNOLDS NUMBER EXPERIMENT

(PART A)

OBJECT

To demonstrate the effect of increasing Reynolds Number on laminar flow.

APPARATUS

A transparent piping system
Constant head tank
Weighing tank and scales
Dye injection mechanism
Stop watch
Thermometer

PROCEDURE

a. Open supply valve sufficiently to maintain a constant head.

b. Check to see that dye system is operating properly.

c. Open discharge valve a small amount. Velocity should be low enough (less than 0.1 ft/sec) to give you laminar flow.

d. Record the time it takes to collect a predetermined weight of water in the weighing tank and the temperature of the water.

e. Using the discharge valve as a control increase the flow rate until transition takes place. Record data at this point.

f. Open discharge valve further to allow you to get turbulent flow. Record data.

EVALUATION OF RESULTS

Explain why your values differ from the accepted Re values.

REYNOLDS NUMBER EXPERIMENT
(PART B)

OBJECT

To determine the effect of varying the Reynolds Number on pressure drop for a straight run of pipe.

APPARATUS

A piping system and manometer
Weighing tank and scales
Stop watch
Thermometer

PROCEDURE

a. Open supply valve wide.

b. Crack open discharge valve until the desired pressure is reached on the manometer (15 - 20 inches of Hg.)

c. Allow a steady state to be reached, and record the time it takes to supply some predetermined weight of water to weighing tank.

d. Record manometer reading and also determine water temperature.

e. Using the discharge valve as a control, adjust the flow rate to nearly closed position taking data for seven different manometer readings.

EVALUATION OF RESULTS

a. Draw curves of manometer pressure (ft of water), mass flow rate (lb/sec), and velocity (ft/sec) as ordinate versus Reynolds Number as abscissa.

b. Referring to the plotted curves, develop an equation for velocity in terms of Reynolds Number.

c. What is the nature of flow? Explain.

DATA (PART "A")

FLOW REGIME	RUN	WEIGHT (lb_f)	TIME (sec)	TEMP. (°F)
LAMINAR	1			
	2			
	3			
	AVE.			
TRANSITIONAL	1			
	2			
	3			
	AVE.			
TURBULENT	1			
	2			
	3			
	AVE.			

PIPE DIAMETER _____ in.

RESULTS (PART "A")

	FLOW REGIME		
	LAMINAR	TRANSITIONAL	TURBULENT
MASS FLOW RATE $\dot{m}$ $\left(lb_m/sec\right)$			
VELOCITY v $\left(ft/sec\right)$			
REYNOLDS NUMBER			

RESULTS (PART "A")

	FLOW REGIME		
	LAMINAR	TRANSITIONAL	TURBULENT
MASS FLOW RATE $\dot{m}$ $\left(lb_m/sec\right)$			
VELOCITY V $\left(ft/sec\right)$			
REYNOLDS NUMBER			

Data Sheet

(Part "B")

P=____psig

Manometer Pressure inches Hg.	Time seconds	Weight pounds

Pipe Diameter _____ in.

Barometer Pressure _____ in. Hg.

Water Temperature _____ °F

Results Sheet

(Part "B")

P= ____ psig

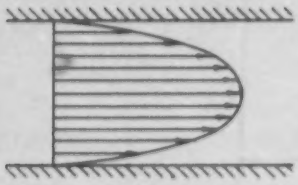
Pressure ft. of H_2O	Mass Flow lb/sec	ϱA lb/ft	Velocity ft/sec	$\varrho D/\mu$ sec/ft	Reynolds No

In laminar flow in ducts and pipes fluid particles move in layers. Each layer moves with uniform velocity in a parallel direction to the particles in the adjacent layers. The particles at the center move at maximum velocity, while away from the center, due to viscous forces, the velocity decreases gradually as the wall is approached. At the wall, a layer of particles adheres to the wall and has zero velocity. At these low Reynolds numbers (Re <2000) the flow is quite orderly and the velocity profile formed by the particles is a paraboloid of revolution with maximum velocity at the center (Figure 1a). When the flow is laminar the average velocity can be calculated in terms of the velocity at the center V_{max}.

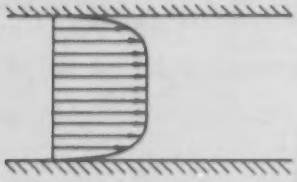
$$V = 1/2\ V_{max}$$

As the Reynolds number increases (Re >3000) the flow becomes turbulent. This type of flow is essentially the one encountered in most actual flow situations. As turbulence develops, the orderly pattern that exists in laminar flow is altered. Small masses of fluids called eddies are formed. These are made up of finite groups of random size traveling in random directions within the main body of fluid. The velocity of these eddies changes with time and distance, and since their formation is a random occurance it is very difficult to treat turbulent flow in the simple analytical fashion that could be used for laminar flow. A consequence of the motion of these particles is a considerable amount of mixing and interchanging of momentum in a traverse direction. As a result of this activity, turbulent flow tends to flatten out the velocity profile creating a more uniform velocity in the pipe except near the wall where the boundary layer exists (Figure 1b).

In this region near the wall, a thin film (laminar sublayer)



(a)



(b)

Figure 1. Velocity Distributions
a. Laminar
b. Turbulent

of laminar flow separates the turbulent region from the wall. The velocity gradient in this region is very high. The particular shape of the velocity profile is useful information. For example, the shear stress at the wall is directly related to the velocity gradient near the wall and since this slope of the velocity profile is greater for turbulent flow it can be seen that the shear stress and losses would be greater for turbulent flow.

The determination of the velocity profile is also often the first step necessary for obtaining the average velocity and discharge through the duct. This is especially the case when instruments such as the pitot tube is the device used. The pitot tube is capable of measuring the velocity pressure only at a point. To obtain the velocity profile it is necessary to traverse the cross section of the duct taking sufficient data to obtain a profile. The average velocity can be determined by one of various area measuring methods and the pitot tube equation (refer to velocity measuring experiment).

$$V = \sqrt{2g\Delta h(\rho_2/\rho_1)}$$

The discharge can be computed from the continuity equation

$$Q = AV$$

where Q = discharge, in cubic feet per second
A = area of duct, in square feet
V = average velocity, in feet per second.

REFERENCES

Binder, R. C., "Fluid Mechanics", Prentice-Hall, Inc., New Jersey, 1962.

Granet, I., "Fluid Mechanics for Engineering Technology", Prentice-Hall, Inc., New Jersey, 1971.

Streeter, V. L., "Fluid Mechanics", McGraw-Hill, 1971.

Swanson W. M., "Fluid Mechanics", Holt, Rinehart and Winston, New York, 1970.

VELOCITY PROFILE EXPERIMENT

OBJECT

To determine the velocity profile and average velocity in a circular duct.

APPARATUS

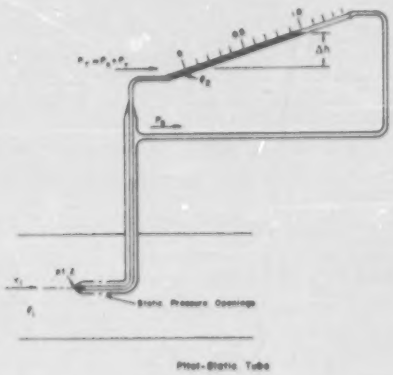
Fan and duct system

Pitot static tube and differential manometer

Strobotac

PROCEDURE

a. Examine pitot tube to make sure it is set up to measure velocity pressure.



b. Set fan at maximum speed.

c. Measure and record the following:
   1. duct diameter
   2. barometric pressure
   3. temperature of air flowing in duct

d. Measure velocity pressure at 1/2 inch increments along the vertical height of the duct.

e. Repeat step (d) for 2/3 maximum speed.

EVALUATION OF RESULTS

a. Plot a graph of velocity in feet per second as abscissa versus diameter position in inches as ordinate (assume zero velocity at the walls).

b. From the graph determine the average velocity by use of a planimeter or other area measuring technique using the following suggested steps:

   1. Find area of the profile in square inches.

   2. Divide the area by diameter of duct in inches. (This results in an abscissa measurement in inch units).

   3. Read corresponding average velocity directly from graph.

c. Determine the relationship between average velocity and V_{max}.

d. Determine the Reynolds number.

e. Determine the volume flow rate in cubic feet per second.

f. Repeat steps (a - e) for the second speed.

Data Sheet

Position (inches)	Velocity Press. (in.)	
	____ rpm	____ rpm
0.0		
0.5		
1.0		
1.5		
2.0		
2.5		
3.0		
3.5		
4.0		
4.5		
5.0		

Sp. Gr. of Manometer Fluid _____

Manometer Correction _____

Barometric Pressure _____

Air Temperature _____

Duct Diameter _____

Results Sheet

Position (inches)	Velocity (ft/sec)	
	_____ rpm	_____ rpm
0.0		
0.5		
1.0		
1.5		
2.0		
2.5		
3.0		
3.5		
4.0		
4.5		
5.0		

Fan Speed (rpm)	Profile Area (sq. in)	Ave. Base (in.)	Ave. Velocity (ft/sec)	$V = KV_{max.}$	Reynolds Number	Capacity (cu ft/sec)
				K=		
				K=		

Flow meters may be classified into two general categories, one measures quantity and the other measures rate of flow. The essential parts of any meter, regardless of its classification, consist of a primary and secondary element. The former (primary) is in direct contact with the fluid and is acted on directly by it; the function of the secondary element is to translate the action of the fluid on the primary element into volumes, weights, or rates of flow and indicate the result. The result may be in the form of a direct quantity measurement or it may be inferred from the effects of the flow rate on pressure, temperature or position.

The orifice plate and venturi are inferential types of metering devices that are widely used to measure flow rates. They have the distinct advantage of being easy to install and in the case of the orifice are inexpensive and readily made. These primary elements are frequently used without a secondary element, and a simple type of manometer is substituted. These head meters (orifices, nozzles, venturis) all use the same basic theory for computing flow.

The primary element is a throttling device placed in a pipe line for the express purpose of creating a measureable difference in pressure between the upstream and downstream sides of the meter. This difference in pressure depends directly on the increase in kinetic energy which is a result of the increase in fluid velocity through the restriction in the meter.

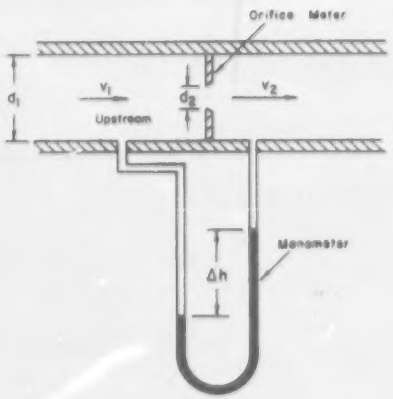


Figure 1. Orifice Meter

(Figure 1 indicates that an orifice plate is being used as the primary element, however, the equations that will be developed may be applied to any head meter).

For incompressible flow, the differential pressure h, in feet, would represent the difference in velocity heads of the fluid flowing through the orifice.

From the Bernoulli equation:

$$h = (V_2^2 - V_1^2)/2g \qquad (1)$$

From the continuity equation:

$$\rho_1 A_1 V_1 = \rho_2 A_2 V_2$$

$$V_1 = V_2 (A_2/A_1) = V_2 (D_2/D_1)^2$$

Then:

$$2gh = V_2^2 - \{V_2(D_2/D_1)^2\}^2$$

$$V_2 = \sqrt{\frac{2gh}{1 - (D_2/D_1)^4}} \qquad (2)$$

Since:

$$Q_1 = Q_2 = A_2 V_2$$

$$Q_{theoret} = A_2 \sqrt{\frac{2gh}{1 - (D_2/D_1)^4}} \qquad (3)$$

Equation 3 is a theoretical equation and will result in the ideal quantity of fluid that will be passed for any given pressure difference. To determine the actual quantity of fluid flowing, it is necessary to determine experimentally a discharge coefficient (C) for the meter in question.

$$Q_{act} = C\, Q_{ideal} \qquad (4)$$

Q_{actual} is the absolute measured quantity of fluid flowing.

Q_{ideal} is the ideal flow of fluid flowing calculated from Equation 3.

C is experimentally determined discharge coefficient.

Therefore

$$Q_{actual} = CA_2\sqrt{\frac{2gh}{1 - (D_2/D_1)^4}} \qquad (5)$$

It is often more convenient to include the diameter ratio as part of the coefficient. The term is called the flow coefficient K, and is defined

$$K = \left[\frac{C}{1 - (D_2/D_1)^4}\right]^{1/2}$$

And $$Q_{actual} = KA_2\sqrt{2gh} \qquad (6)$$

An examination of equation (6) indicates for a constant head (h) the flow (Q) is proportional to the area (A).

In the variable area meters, and the rotameter falls in this category, the head (or pressure drop) is constant and the flow rate is indicated as a function of the area.

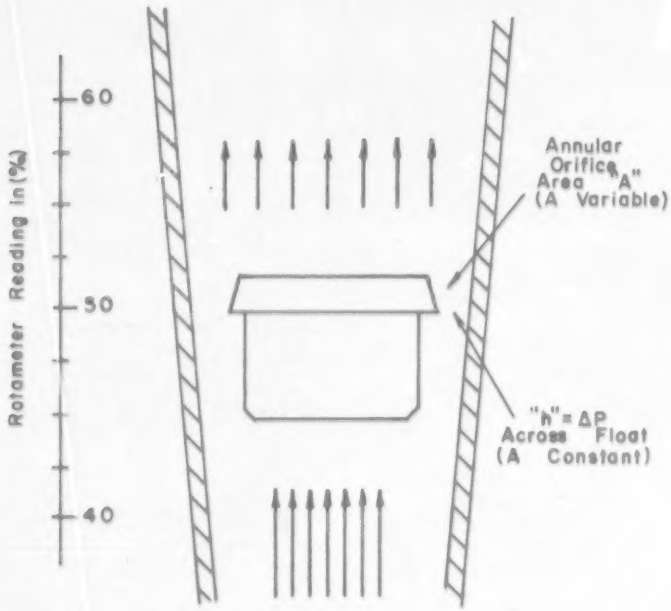


Figure 2. Rotameter

As shown in diagram the scale reading is given in percent, and the quantity of fluid flowing is determined directly by multiplying the design flow rate by the reading indicated at the sharp edge of the float.

REFERENCES

Ambrosius, E. E., Fellows, R. D., and Brickman, A. D., "Mechanical Measurements and Instrumentation", Ronald Press, Inc., N.Y., 1966.

O'Higgins, P. J., "Basic Instrumentation, Industrial Measurement", McGraw-Hill, N. Y., 1966.

Tuve, G. L., "Mechanical Engineering Experimentation", McGraw Hill, N. Y., 1961.

Variable Area Flowmeter Handbook, Volume I - Basic Rotameter Principles, Fischer-Porter, 1968.

Therefore

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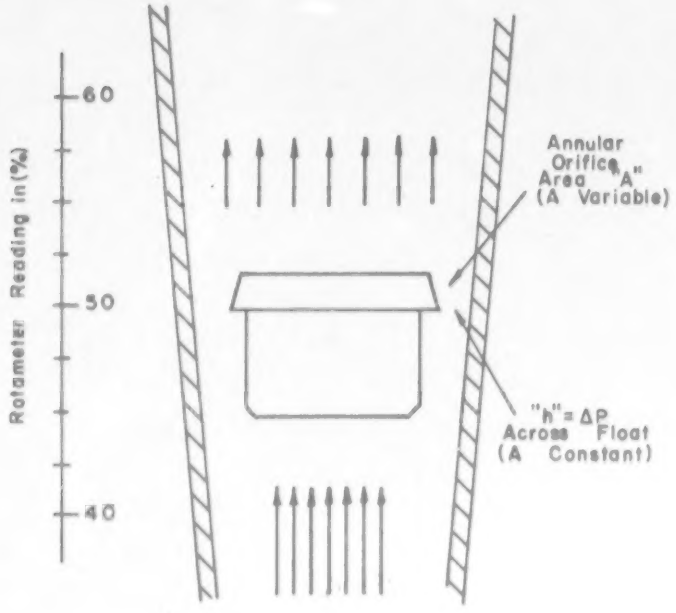

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Tuve, G. L., "Mechanical Engineering Experimentation", McGraw Hill, N. Y., 1961.

Variable Area Flowmeter Handbook, Volume I - Basic Rotameter Principles, Fischer-Porter, 1968.

FLOW METER COEFFICIENTS

OBJECT

To determine experimentally the discharge coefficients of an orifice plate, venturi and rotameter at varying flow rates.

APPARATUS

Orifice Plate
Venturi Meter
Rotameter
Thermometer
Weighing Tanks
Differential Manometers
Stop Watch

PROCEDURE

a. Examine flow meter arrangement and check manometers to see if they are properly connected to tap-offs and are at zero difference.

b. Fill up the tank with water, and with the control valve closed, start pump.

c. After making sure that manometers indicate zero pressure difference, slowly open control valve until maximum pressure drop is obtained across the meters.

d. When steady state has been attained, allow a predetermined weight of water to enter weighing tanks and record the time it takes to complete this process.

e. Record the pressure drop across the venturi, orifice plate, the reading on the rotameter and the water temperature.

f. Using the discharge valve as a control, reduce the flow rate by intervals of 1/2 gallons per minute (as indicated on rotameter). Record data for new flow rate.

EVALUATION OF RESULTS

a. Determine discharge coefficients for each device at all flow rates.

b. Plot discharge coefficient for each meter against ideal flow rate.

c. The relationship between discharge and pressure drop for a particular head meter (orifice or venturi) is given by the following relationship $Q \propto \sqrt{\Delta h}$ (discharge is proportional to the square root of the pressure drop). Verify from an investigation of data and results the accuracy of this relationship.

Data Sheet

Run Number	Weight of Water (lb)	Time (sec)	Pressure Drop		Rotameter Reading (gpm)	Water Temperature (°F)
			Orifice (in.)	Venturi (in.)		
1						
2						
3						
4						
5						
6						
7						
8						
9						
10						

Pipe Diameter _____

Venturi Throat Diameter _____

Orifice Diameter _____

Results Sheet

Run Number	Actual Flow			Ideal Flow (cfs)			Coefficient "C"		
	$\dot{m}$ (lb/sec)	V (fps)	Q (cfs)	Orifice	Venturi	Rotameter	Orifice	Venturi	Rotameter
1									
2									
3									
4									
5									
6									
7									
8									
9									
10									

Results Sheet

Run Number	Actual Flow			Ideal Flow (cfs)			Coefficient "C"		
	$\dot{m}$ (lb/sec)	V (fps)	Q (cfs)	Orifice	Venturi	Rotameter	Orifice	Venturi	Rotameter
1									
2									
3									
4									
5									
6									
7									
8									
9									
10									

LOST HEAD IN PIPES AND FITTINGS

EXPERIMENT NO. 13

The flow of fluids through pipes and fittings may be either laminar or turbulent. Regardless of the flow regime, certain losses are inevitable due to friction and will result in a pressure drop within any piping system. When laminar flow exists, the friction loss is directly proportional to the velocity. However, if flow is turbulent, the losses are nearly proportional to the square of the mean velocity.

For most engineering applications the flow regime is turbulent, and the basic equation for determination of head loss in a uniform length of straight pipe is given by the following equation:

$$h_1 - h_2 = f \frac{L}{D} \frac{V^2}{2g}$$

L = length of pipe in feet
V = mean fluid velocity in feet per sec.
D = inside pipe diameter in feet
f = friction factor (a function of Reynolds Number (R) and relative roughness) from Moody diagram
h_1-h_2 = head loss in feet of fluid

In cases where flow regime is laminar, f is experimentally determined to be 64/R, and the basic equation for head loss becomes:

$$h_1 - h_2 = \frac{64}{R} \frac{L}{D} \frac{V^2}{2g}$$

When a piping system contains valves and fittings, the L/D ratio has no significance as far as the valves and fittings are concerned and a dimensionless coefficient "K" must be calculated for them. This coefficient, referred to as an interference coefficient, is determined by the following equation.

$$h_1 - h_2 = K \frac{V^2}{2g}$$

$h_1 - h_2$ = head loss due to fitting in feet of fluid

K = interference coefficient (typical coefficients are given in following table)

V = mean fluid velocity in feet per second

INTERFERENCE COEFFICIENTS FOR VALVES AND FITTINGS

VALVE	K	FITTING	K
Globe valve - wide open	10.0	Return bend	2.2
Angle valve - wide open	5.0	Standard tee	1.8
Gate valve - wide open	0.19	Standard elbow	0.9
Gate valve - ½ closed	5.6	45° elbow	0.42
Gate valve - ¼ closed	1.15	Long sweep elbow	0.60
Swing - Check - Open	2.5	Medium sweep elbow	0.75

Another technique of expressing the losses in a piping system is to convert the components of the system (fittings, valves, etc.) into equivalent lengths of straight pipe from available tables. The equivalent length is then added to the existing lengths of straight pipe. This total length can now be multiplied by table values of loss per foot resulting in the total head loss for the piping system.

Mathematically the equivalent length for each component may be computed by the following equation:

$$L = \frac{DK}{f}$$

f = friction factor for straight pipe

D = inside pipe diameter in feet

K = interference coefficient

L = equivalent length of straight pipe for particular fitting

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LOST HEAD IN PIPES AND FITTINGS EXPERIMENT

OBJECT

To find the friction factor "f" for smooth copper at various flow rates.

To find the interference coefficient "K" for various fittings at each flow rate.

APPARATUS

A piping system with fittings and manometers weighing tanks and scales.
Stop watch
Thermometer

PROCEDURE

a. Trace out piping system and complete piping diagram.

b. Open supply valve wide.

c. Crack open discharge valve until maximum allowable pressure drop is reached on manometer #6 (across globe valve).

d. Allow a steady state to be reached, and record the time it takes to supply some predetermined weight of water to weighing tanks.

e. Record manometer readings for each station and also determine water temperature.

f. Using the discharge valve as a control, adjust flow rate for three more runs at approximately 1/4, 1/2 and 3/4 of maximum flow rate. Take all data.

EVALUATION OF RESULTS

a. Draw a curve using log paper (1 x 1) of friction factor "f" as ordinate versus Reynolds Number as abscissa.

b. Average the values of interference coefficient for all flow rates and compare with values from reference material.

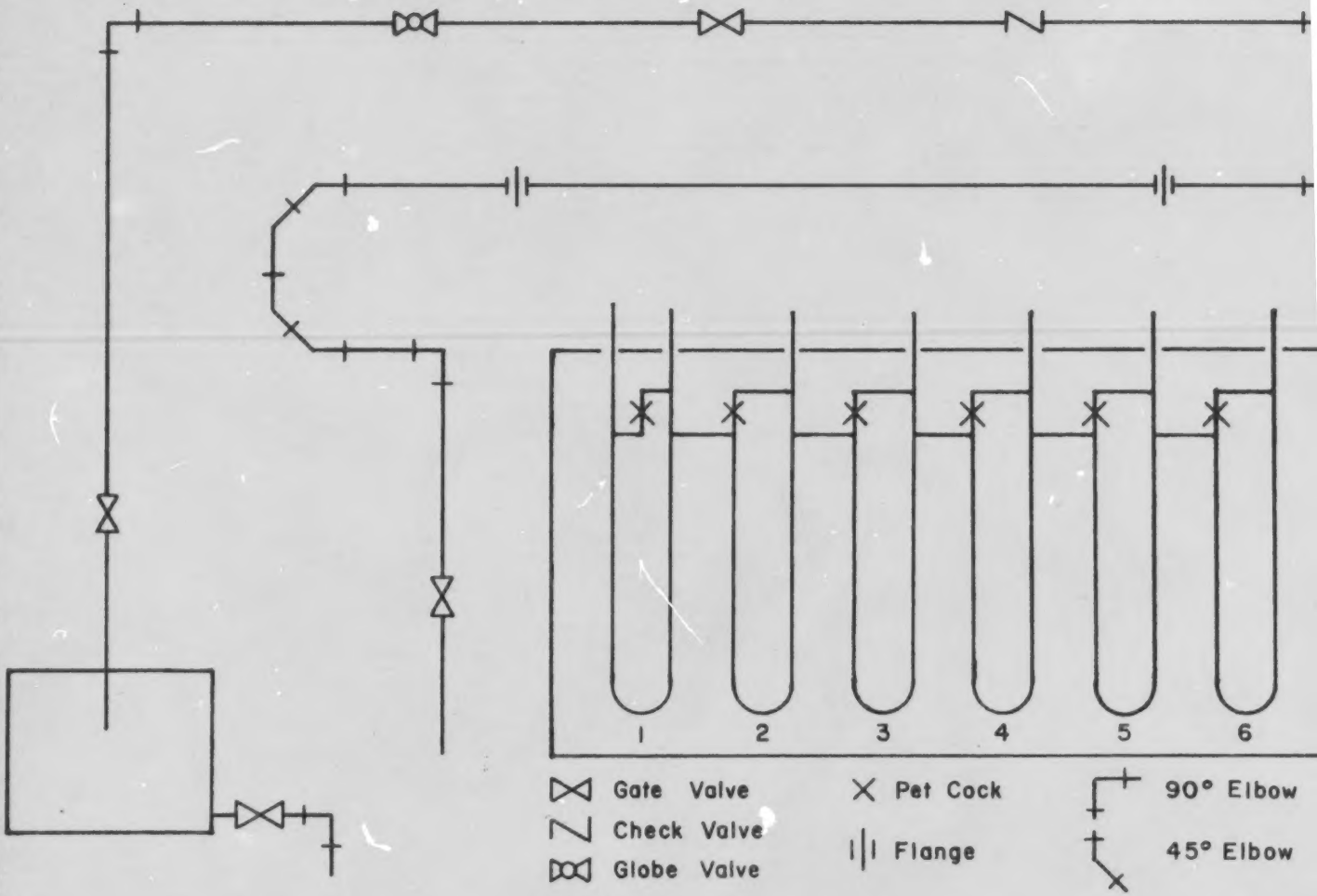

Figure I. Lost Head Piping System

Data Sheet

Sta. No.	Fittings	Manometer Readings (inches of Hg)			
		1	2	3	4
1	4-45° elbows & 7.53ft of pipe				
2	14.73ft of pipe				
3	2-90° elbows & 8.75ft of pipe				
4	Check Valve & 5.48ft of pipe				
5	Gate Valve & 5.48ft of pipe				
6	Globe Valve & 5.56ft of pipe				
	Time (seconds)				
	Water Temperature				
	Weight of water				

Pipe Diameter _____ feet

Results Sheet

Test		1	2	3	4
Mass-Flow-Rate lb/sec					
Flow Velocity fps					
Reynolds Number					
Lost Head ft of water	14.73 ft pipe				
	per foot of pipe				
Lost Head - feet of water	45° elbow				
	90° elbow				
	Check Valve				
	Gate Valve				
	Globe Valve				
Interference Coefficient K	1 - 45° elbow				
	1 - 90° elbow				
	Check Valve				
	Gate Valve				
	Globe Valve				
Frict. Factor	14.73 ft pipe				

A centrifugal pump is designed to convert the kinetic energy of a flowing stream into pressure energy and to transport a fluid from a region of lower pressure to a region of higher pressure. The pump consists of one or more impellers mounted on a rotating shaft and revolves in an enclosed casing. The shaft is normally coupled to a direct drive of some nature, usually a motor or turbine.

The casing is filled with the fluid to be pumped before starting to prevent the pump from being air bound (air trapped in the casing). The impeller is set in motion and fluid enters the impeller in an axial direction near the shaft and the rotating vanes of the impeller throw the fluid outward in the casing imparting both kinetic and potential energy to the fluid. This high velocity fluid is collected in a volute or series of diffusing passages which converts the kinetic energy of the fluid to pressure energy.

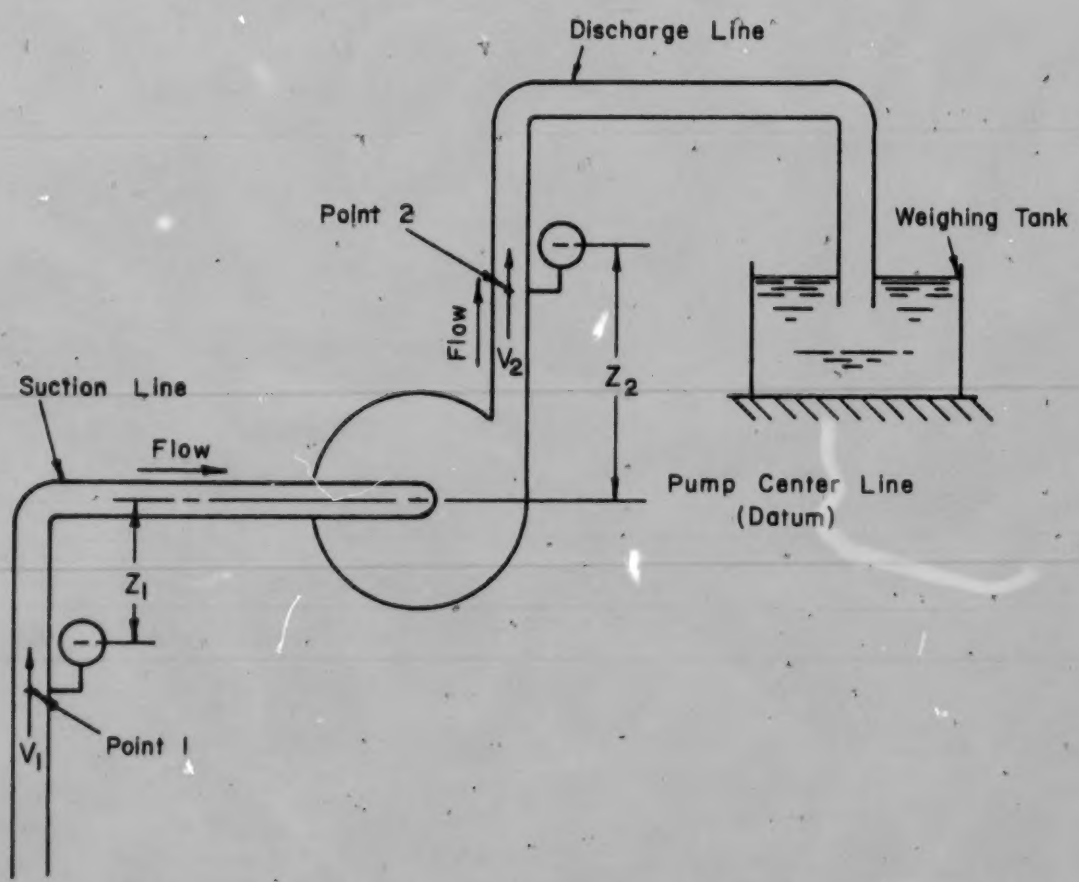

Figure 1. Centrifugal Pump Arrangement

Applying the steady flow equation between points 1 and 2 in units of Btu/lbm.

$$PE_1 + KE_1 + Ps_1/\rho_1 J + U_1 + {}_1Q_2 = PE_2 + KE_2 + Ps_2/\rho_2 J + U_2 + {}_1W_2 + E_{loss(1\rightarrow 2)}$$

$$gZ_1/g_c J + V_1^2/2g_c J + Ps_1/\rho_1 J + U_1 + {}_1Q_2 = gZ_2/g_c J + V_2^2/2g_c J + U_2 + {}_1W_2 +$$

$$E_{loss(1\rightarrow 2)} \qquad (2)$$

PE = potential energy
KE = kinetic energy
g_c = gravitational constant = 32.2 lbm-ft/lbf-sec^2
g = local acceleration of gravity
J = Joule's constant 778 ft-lbf/Btu
Z = elevation distance from center line of pressure gage to center line of pump (feet)
V = average velocity of fluid (ft/sec)
Ps = absolute static pressure (p.s.f.g.)
ρ = fluid density (lbm/ft^3)
U = internal energy of fluid (Btu/lbm)
Q = net transferred heat (Btu/lbm)
W = net work between points 1 and 2 (Btu/lbm)
E_{loss} = energy loss due to fittings and pipe friction between points 1 and 2

Although there are mechanical and hydraulic losses in the pump, these are taken care of in determining the efficiency of the pump.

Assuming the flow to be adiabatic (${}_1Q_2 = 0$) and the changes in temperature between points 1 and 2 to be negligible ($U_1 = U_2$), equation (2) becomes

$$gZ_1/g_c J + V_1^2/2g_c J + Ps_1/\rho_1 J = gZ_2/g_c J + V_2^2/2g_c J + Ps_2/\rho_2 J + {}_1W_2 +$$

$$E_{loss(1\rightarrow 2)} \qquad (3)$$

To convert equation 3 into units of ft-lbf/lbm

a. Multiply both sides of equation by $g_c J/g$
b. Let gamma $\gamma = \rho g/g_c$
c. Assume incompressible flow with $\rho_1 = \rho_2$
d. $g = g_c$ numerically; therefore 1 lbf = 1 lbm

Therefore:

$$Z_1 + V_1^2/2g + Ps_1/\gamma = Z_2 + V_2^2/2g + Ps_2/\gamma + J_1W_2 + JE_{loss(1\rightarrow 2)} \qquad (4)$$

Each term in equation (4) has units of ft-lbf/lbm or feet of fluid flowing.

$h_v = V^2/2g$ is termed the velocity head.
$h_s = Ps/\gamma$ is termed the static head.
J_1W_2 represents the pump work in units of ft-lbf/lbm.
$JE_{loss(1\rightarrow 2)}$ represents the energy losses in ft-lbf/lbm.

The JW term is referred to as total head (H_T) the $JE_{Loss\ (1-2)}$ is called the friction head (h_f).

Substituting these values in the previous equation and solving for the total head (H_T), we find that:

$$H_T = (h_{s1} - h_{s2}) + (h_{v1} - h_{v2}) + (Z_1 - Z_2) - h_f \qquad (5)$$

Analyzing the equation in reference to Figure 1, the H_T will be negative ($hs_2 > hs_1$). This indicates that the pump is doing work on the fluid.

Further simplification of equation (5) can be done by defining the following:

$H_{TS} = h_{s1} + h_{v1} + Z_1$ total suction head
$H_{TD} = h_{s2} + h_{v2} + Z_2$ total discharge head

Therefore:

$$H_T = H_{TS} - H_{TD} - h_f \qquad (6)$$

In practical work and many piping systems the value of h_f is significant and must be considered. For this experiment the location of the pressure measuring stations are close to the pump, eliminating large lengths of pipe and fittings, therefore, the value of h_f will be considered negligible.

Hydraulic horsepower or water horsepower is expressed as:

$$HP = \frac{\dot{W}H_T}{550} \qquad (7)$$

Where $\dot{W}$ is the weight flow rate lbf/sec
H_T is the total head ft.

This is the actual useful horsepower delivered by the pump unit.

The power input to the motor is calculated from the electric potential (in volts) across the motor and the current (in amps) through the motor.

$$W_{(\text{in motor})} = \frac{(\text{Volts})\ (\text{Amps})\sqrt{3}}{1000} \quad \text{KW} \qquad (8)$$

The power input to the pump is the brake horsepower of the driving motor. This may be calculated by referring to manufacturer's performance curve of motor efficiency versus KW input.

$$BHP_{(\text{in pump})} = W_{(\text{in motor})}\ \text{KW} \times \text{power factor} \times \text{Eff.}\left(1.341\ \frac{HP}{KW}\right) \qquad (9)$$

In some cases data may be available relating motor watts input to overall efficiency of motor and drive. The relationship for the brake horsepower to the pump can be simplified to:

$$BHP_{(\text{in pump})} = W_{(\text{in motor})}\ \text{KW} \times \text{Eff.}_{(\text{motor \& drive})}\left(1.341\ \frac{HP}{KW}\right) \qquad (10)$$

A pump's performance may be expressed in terms of how efficiently it uses energy supplied to it - what percentage of the energy imparted to the pump by the motor is transmitted to the water? In terms of an equation the efficiency of a pump may be expressed as a ratio of the two horsepowers.

$$\eta_{pump} = \frac{HHP}{BHP} \qquad (11)$$

In industrial practice a term used in classifying pumps is specific speed (n_s).

$$n_s = \frac{N\sqrt{Q}}{(H_T)^{3/4}} \qquad (12)$$

Where:

Q = capacity in gallons per minute
H_T = total head per stage in feet
N = revolutions per minute

The values of H_T and Q for substitution in the above equation are the values at maximum efficiency for the selected revolutions per minute. By knowing the value of specific speed, an impeller can be designed or a pump can be chosen for optimum performance.

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CENTRIFUGAL PUMP EXPERIMENT

OBJECT

To operate a centrifugal pump and determine its characteristic curves.

APPARATUS

Centrifugal pump with motor-drive
Weighing tanks
Stop watch
Strobotac
Voltmeter and Ammeter

PROCEDURE

a. Make a block diagram of the pump and the associated piping. Identify both suction and discharge gages and the dimensions that indicate their height above or below the pump centerline.

b. Open city water line to inlet side of pump to prime pump.

c. Open discharge valve wide and start pump; then close city water line.

d. Close discharge valve and note pressure reading.

e. Open discharge valve wide and note pressure reading. The difference in these two values will determine the pressure range for the operating speed of the pump.

f. Subdivide this range into 5 - 6 readings (eliminating shut-off pressure).

g. Run first test with discharge valve wide open and pump two hundred pounds of water into weighing tanks. Record all data.

h. Repeat this procedure for the ramaining valve settings.

i. Calculate for each discharge valve setting:

1. Total discharge head, H_{TD}
2. Total suction head, H_{TS}
3. Total head, H_T
4. Hydraulic horse power, HHP
5. Input to motor, KW
6. Input horsepower to pump, HP
7. Pump efficiency, η
8. Capacity in gallons per minute
9. Specific speed, n_s

EVALUATION OF RESULTS

a. Plot the following curves with capacity (gpm) as abscissa.

1. Total head
2. Brake horsepower
3. Pump efficiency

b. Discuss completely the test results as shown by the curves.

Data Sheet

At _____ rpm

Discharge Pressure (psig)	Suction Pressure (in. Hg)	Volts	Amps	Weight (lbs.)	Time (sec.)

Pipe Sizes: Inlet Diameter _____

Outlet Diameter _____

Gage Location from Pump Center Line:

Barometric Pressure _____

Water Temperature _____

Discharge Line _____

Suction Line _____

Results Sheet

Total Disch. Head (feet)	Total Suct. Head (feet)	Total Head (feet)	Water Horsepower	Motor			Pump Efficiency	Capacity (gal/min)
				KW Input	Efficiency* η	HP Output		

Specific Speed _____

* Efficiency of Vari-Drive Unit including Power Factor

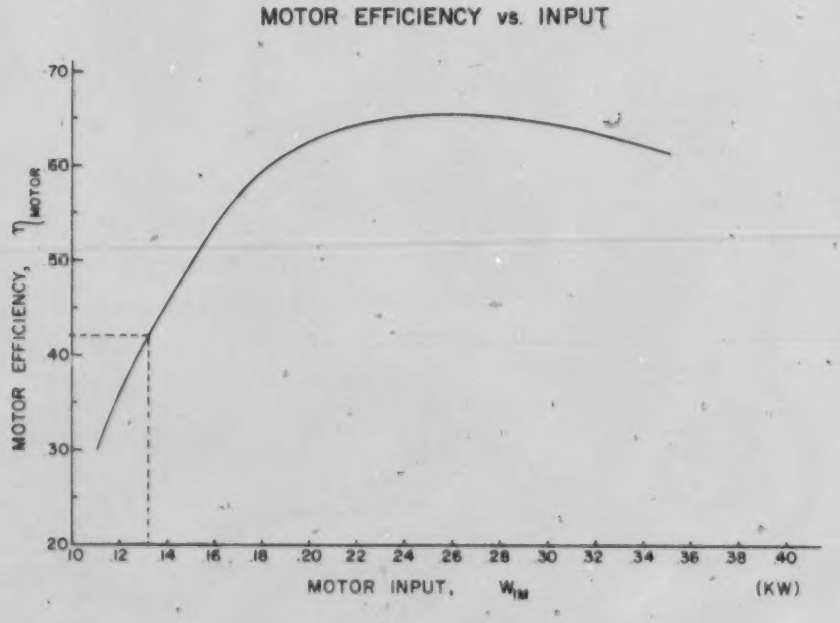
Fans and blowers for the controlled circulation of air have extensive applications in the design of power plants and heating and ventilating systems. Regardless of the design, whether the fan is axial flow or centrifugal the technique of determining their respective operating characteristics is the same. However, there is a vast difference in their applications and the selection of the proper fan for a particular system depends on many variables.

A fan essentially is an air mover, a machine which uses energy obtained from an electric motor (the fan input), to transport a quantity of air (the fan output). The fan's performance may be expressed in terms of how efficiently it uses energy supplied to it - what percentage of the energy imparted to the fan by the motor is transmitted to the air?

The power input to the motor is calculated from the electric potential (in volts) across the motor and the current (in amps) through the motor.

$$W_{(\text{in motor})} = \frac{(\text{Volts})\ (\text{Amps})}{1000}\ \text{KW}$$

The power input to the fan is the brake horsepower of the driving motor. This may now be calculated by referring to manufacturer's performance curve of motor efficiency versus KW input such as shown in the following example:



$$W_{(\text{in fan})} = W_{(\text{in motor})}\ KW(\eta_{\text{motor}})\ 1.341\frac{HP}{KW}$$

The fan moves a quantity of air against a pressure increase (fan inlet to fan outlet.) The output power of the fan is the amount of work done by the fan in moving the air through the duct. The pressure increase ΔP may be expressed in terms of the height of a column of fluid, and in this case, we let it be a column of air, h_A, the same density as that flowing through the duct. The fan output power therefore is the amount of energy necessary to raise this same amount of air, at the same rate, a height equal to the length of the column of air representing ΔP.

The rest of the material in this experiment deals with the various quantities necessary to calculate the output power, volume rate of flow and efficiency of the fan.

The density of the air in the duct is a function of pressure and temperature. It may be read off a chart or table, or it may be calculated with the equation of state.

$$\rho_A = \frac{P}{RT}$$

P = absolute static pressure	lbf/ft^2
R = gas constant, for air 53.35	$\frac{ft\text{-}lbf}{lbm\text{-}^oR}$
T = absolute air temperature	

Where the absolute static pressure equals

$$P_A = P_S + \text{Barom. Press}$$

Fan testing set-ups generally are instrumented with either a pitot tube or orifice plate. Most fan testing in the field is done with a pitot-static tube. This device enables one to measure the static pressure, total pressure, or the difference of these two, velocity pressure at a point. To obtain average values it is necessary to traverse at least along two diameters of the duct.

In laboratory set-ups it may be more convenient to use the orifice plate. The orifice is a restriction, through which passes the total mass of air. The static pressure differential across the orifice is a function of the average velocity through it. The use of the orifice plate eliminates the need of traversing.

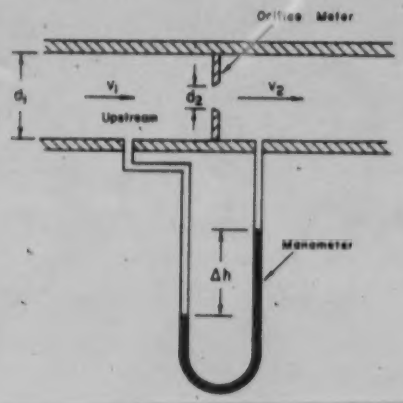


Figure 1. Orifice Meter

Figure 1 shows the metering arrangement used in this experiment. The static pressure is measured upstream from the orifice, at a distance of one duct diameter from the face of the orifice plate. The downstream static pressure is measured at a distance of one-half a duct diameter from the downstream face of the plate. The relationship for the velocity can be derived by applying Bernoulli's equation to a streamline passing through the center of the orifice. The following is the resulting form of the equation generally used to find the average velocity in the duct.

$$V = 1096(d/D)^2 \; C_d\sqrt{\frac{\Delta h}{\rho_A}}$$

V = air velocity in duct	ft/min
d = orifice diameter	in.
D = duct diameter	in.
C_d = discharge coefficient	
Δh = manometer reading	in. H_2O
ρ_A = density of air in duct	lbm/ft^3

The discharge coefficient of the orifice, C_d, depends on the ratio of the orifice diameter to the duct diameter, and is read off a chart. In this particular example using a 5.75 inch diameter duct, and orifice to duct diameter ratio, d/D, of 0.6, we obtain an orifice coefficient, C_d, of 0.665.

To calculate the drop in static pressure between the fan outlet and the upstream pressure tap, it is necessary to obtain the velocity pressure.

$$P_V = (d/D)^4 \, C_d^{\,2} \, \Delta h$$

P_V = velocity pressure in duct	in. H_2O
Δh = manometer reading	in. H_2O

The static pressure drop can be calculated from the following equation:

$$\Delta P_S = 0.02 \, \frac{L}{D} \, P_V$$

ΔP_S = static pressure drop	in. H_2O
L = length of duct between fan outlet, and upstream pressure tap	in.
D = duct diameter	in.
P_V = velocity pressure in duct	in. H_2O

This allows us to obtain the static pressure at the fan outlet.

$$P_{Sfo} = P_S + \Delta P_S$$

P_{Sfo} = static pressure at fan outlet	in. H_2O
P_S = static pressure at upstream tap	in. H_2O
ΔP_S = static pressure drop in duct	in. H_2O

The total pressure is the sum of the static and velocity pressures.

$$P_T = P_S + P_V$$

P_T = total pressure in the duct	in. H_2O
P_S = static pressure at the upstream tap	in. H_2O
P_V = velocity pressure	in. H_2O

Using the continuity equation we obtain volume and mass flow rates.

$$Q = VA$$

Q = volume rate of flow	ft^3/min
V = air velocity in duct	ft/min
A = cross sectional area in duct	ft^2

$$\dot{m} = \rho_A Q$$

$\dot{m}$ = mass rate of flow	lbm/min
ρ_A = density of air in duct	lbm/ft^3
Q = volume rate of flow	ft^3/min

The total and static fan outlet pressures can now be calculated.

$$h_{AT} = \frac{(P_T) \text{ in } H_2O \ (\rho H_2O)}{12 \text{ in/ft } (\rho air)}$$

$$h_{AT} = \frac{5.2 \ P_T}{\rho_A}$$

$$h_{AS} = \frac{5.2 \ P_{Sfo}}{\rho_A}$$

h_{AT}	= fan outlet total pressure	ft-air
h_{AS}	= fan outlet static pressure	ft-air
P_T	= fan outlet total pressure	in. H_2O
P_{Sfo}	= fan outlet static pressure	in. H_2O
ρ_A	= density of air in duct	lbm/ft^3

Once the static and total pressures have been determined the static and total horsepower can be calculated from the following equation:

$$HP_{AS} = \frac{\dot{m}g \ h_{AS}}{g_c \ 33{,}000 \ \frac{\text{ft-lbf}}{\text{min-HP}}}$$

$$HP_{AT} = \frac{\dot{m}g \ h_{AT}}{g_c \ 33{,}000 \ \frac{\text{ft-lbf}}{\text{min-HP}}}$$

HP_{AS}	= static air horsepower	HP
HP_{AT}	= total air horsepower	HP
$\dot{m}$	= mass rate of flow	lbm/min
h_{AS}	= fan outlet static pressure	ft-air
H_{AT}	= fan outlet total pressure	ft-air

The ratio of these air horsepowers to the input horsepower, gives us the efficiencies of the fan.

$$\eta_{FS} = \frac{HP_{AS}}{W_{om}}$$

$$\eta_{FT} = \frac{HP_{AT}}{W_{om}}$$

η_{FS} = static fan efficiency

η_{FT} = total fan efficiency

HP_{AS} = static air horsepower HP

HP_{AT} = total air horsepower HP

W_{om} = output from motor (fan input) HP

As can be seen, fan outlet pressures, horsepowers and efficiencies are evaluated both in static and total terms. The results obtained by use of total pressure data are considerably more significant. It is however, necessary to deal with static values since most fan manufacturers tend to include these in their performance characteristics.

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FAN TESTING EXPERIMENT

OBJECT

To determine the operating characteristics of a centrifugal fan running at its rated speed.

APPARATUS

Test set-up
Strobatac

PROCEDURE

a. Examine fan to make sure that it is in good condition and that the bearings are lubricated.

b. Determine rated fan speed.

c. Set fan at maximum speed with throttling device at duct exit in wide open position. Maintain this speed for all runs.

d. Once flow has stabilized record the pressure drop across the orifice and the upstream static pressure.

e. Record temperature of air flowing in duct.

f. In addition to above also record volts and amps input to fan motor.

g. With fan operating at selected speed, adjust throttling device to 0.9, 0.8, 0.7 ..., and fully closed position and for each setting repeat steps d to f as outlined.

h. Record the rest of the information called for on the data sheet.

EVALUATION OF RESULTS

a. Plot on rectangular coordinates the horsepower input to fan, the total and static efficiency, the total and static pressure as ordinate versus the capacity in cubic feet minute.

b. Compare the results against data presented in manufacturers catalogs for same type of fan.

DATA

SPEED OF FAN = RPM

DUCT EXIT SETTING	MOTOR INPUT		AIR TEMP.	MANOMETER READING		CORRECTED MAN. RDG.*	
				ACROSS ORIFICE	UPSTREAM STATIC PRESS.	ACROSS ORIFICE Δh	UPSTREAM STATIC PRESS. P_s
	VOLTS	AMPS	°F	IN H_2O	IN H_2O	IN H_2O	IN H_2O
OPEN							
.9							
.8							
.7							
.6							
.5							
.4							
.3							
.2							
.1							
CLOSED							

* CORRECTED MAN. RDG. = MANOMETER READING X MANOMETER SCALE FACTOR

ORIFICE DIAMETER, d = IN ORIFICE COEFFICIENT, Cd =

DUCT DIAMETER, D = IN BAROMETRIC PRESSURE = IN Hg

LENGTH OF DUCT BETWEEN FAN OUTLET AND UPSTREAM PRESSURE TAP, L = IN

MANOMETER SCALE FACTORS: UPSTREAM PRESSURE TAP =

ACROSS ORIFICE =

RESULTS

FAN SPEED = RPM

DUCT EXIT SETTING	MOTOR			AIR VELOCITY	STATIC PRESS. DROP	FAN OUTLET PRESSURES	
	INPUT W_{IM}	EFF. η_{MOTOR}	OUTPUT W_{OM}	V	Δp_s	STATIC p_{sfo}	TOTAL p_t
	KW		HP	FT/MIN	$"H_2O$	$"H_2O$	$"H_2O$
OPEN							
0.9							
0.8							
0.7							
0.6							
0.5							
0.4							
0.3							
0.2							
0.1							
CLOSED							

	AIR RATE OF FLOW		FAN OUTLET PRESSURES		FAN AIR HORSE POWER		FAN EFFICIENCIES	
	VOLUME Q	MASS $\dot{m}$	STATIC h_{AS}	TOTAL h_{AT}	STATIC HP_{AS}	TOTAL HP_{AT}	STATIC η_{FS}	TOTAL η_{FT}
	FT^3/MIN	LBM/MIN	FT-AIR	FT-AIR	HP	HP	%	%
1.0								
0.9								
0.8								
0.7								
0.6								
0.5								
0.4								
0.3								
0.2								
0.1								
0								

FAN LAWS EXPERIMENT NO. 16

At times it is necessary to determine the effect of a speed change on the capacity, total head, and horsepower input of a given fan. The result of this speed change can be predicted by application of the fan laws to fans having the same geometric shape and operating at the same point of rating on the performance curve.

Capacity:

From the continuity equation for incompressible flow

$$Q = AV$$

Where Q = capacity in $ft^3/min.$

A = Area in ft^2

V = Average velocity in ft/min.

Consider the area of the impeller as

$$A = \pi/4\ d^2$$

and the peripheral speed as

$$V = \pi dN$$

Where d = impeller diameter in feet

N = revolutions per minute of impeller

Therefore:

$$Q = 0.7854\ d^2\ (\pi dN)$$

Although the results would be approximate, the capacity varies directly with the speed for a given fan diameter, and:

$$Q_1/Q_2 = N_1/N_2 \qquad (1)$$

Head:

The pressure or head developed by a fan is a function of the kinetic energy of the impeller.

$$H_T \propto V^2/2g$$

Therefore: $$H_T \propto (\pi dN)^2/2g$$

$$H_{T_1}/H_{T_2} = (N_1/N_2)^2 \qquad (2)$$

Brake horsepower:

$$BHP = AHP/\eta \tag{3}$$

$$AHP = \dot{m}H_T/33{,}000 \tag{4}$$

$$\dot{m} = \gamma Q \tag{5}$$

Therefore:

$$AHP = \gamma Q_1 H_T/33{,}000 \tag{6}$$

or

$$AHP_1/AHP_2 = Q_1 H_{T_1}/Q_2 H_{T_2} \tag{7}$$

Since

$$Q_1/Q_2 = N_1/N_2$$

and

$$H_{T_1}/H_{T_2} = (N_1/N_2)^2$$

Substituting into equation (7) the following relationship is developed:

$$AHP_1/AHP_2 = (N_1/N_2)^3 \tag{8}$$

For the same point of efficiency:

$$BHP_1/BHP_2 = (N_1/N_2)^3 \tag{9}$$

Equation 1, 2 and 9 are known as the fan laws as applied to changes in fan speed.

1. The capacity varies as fan speed.
2. The total head varies as fan speed squared.
3. The brake horsepower varies as fan speed cubed.

(Although the preceeding information was applied directly to fans, the basic theory may be applied to centrifugal pumps).

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Church, A. H., "Centrifugal Pumps and Blowers", John Wiley and Sons, Inc., New York, 1960.

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FAN LAW EXPERIMENT

OBJECT

To investigate the effect of varying fan speed on capacity, total head and brake horsepower input.

APPARATUS

Fan test set-up

Strobotac

PROCEDURE

a. Set fan at predetermined speed with duct exit fully open.

b. Record the following data at this fan speed.

   1. Fan speed
   2. Upstream static pressure (total pressure for pitot tube)
   3. Static pressure difference (total pressure for pitot tube)
   4. Volts and Amps to fan motor

c. Decrease fan speed in increments of 200 revolutions per minute and record data as specified in step "b" for each speed.

d. Calculate KW inputs to fan and pick off from efficiency versus load curve the proper efficiency.

EVALUATION OF RESULTS

a. Determine the capacity in ft^3/min and brake horsepower input for each speed and record on result sheet along with the total head in inches of water.

b. Plot on log-log paper; capacity, total head and brake horsepower input as ordinates versus speed as abscissa.

c. Determine the slope of the three curves.

d. Discuss any discrepencies that exist between the slope of the plotted curves and the expected shape.

FAN LAW EXPERIMENT

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d. Discuss any discrepencies that exist between the slope of the plotted curves and the expected shape.

Data Sheet

(Pitot - Tube)

Fan Speed rpm	Manometer Readings (H_2O)		Motor Input			Motor Eff. (%) ♣
	Total	Static	Volts	Amps	KW	

Air Temperature _____ Barometric Pressure _____

Manometer No. 1 Manometer No. 2

Initial Reading _____ Initial Reading _____

Correction Factor _____ Correction Factor _____

♣ from efficiency curve

Results Sheet

(Pitot - Tube)

Fan Speed RPM	Corrected Manometer Readings ("H_2O)			Velocity (fpm)	Capacity (cfm)	Total Head (ft of Air)	Input to Fan (HP)
	Total	Static	Velocity				

Duct Area _____ Air Density _____

Data Sheet

(Orifice)

Fan Speed rpm	Manometer Readings ($"H_2O$)		Motor Input			Motor Eff. (%) *
	Upstream Static	Static Diff.	Volts	Amps	KW	

Orifice Diameter _____

Duct Diameter _____

Orifice Coefficient _____

Air Temperature _____

Barometric Pressure _____

* from efficiency curve

Manometer No. 1

Initial Reading _____

Correction Factor _____

Manometer No. 2

Initial Reading _____

Correction Factor _____

Results Sheet

(Orifice)

Fan Speed (rpm)	Static Diff. (in. of H_2O)	Velocity (ft/min)	Capacity (cu ft/min)	Velocity Head (in. of H_2O)	Static Head (in. of H_2O)	Total Head (in. of H_2O)	Input to Fan (Horsepower)

Duct Area _____ Air Density _____

The demand for the most economical method of producing power from internal combusion engines requires that automotive engineers and technicians have a complete knowledge of the techniques involved in engine testing. A variety of tests may be performed on any one engine, but a thorough investigation of all of these is beyond the scope of this discussion. An examination of the following areas will be considered :

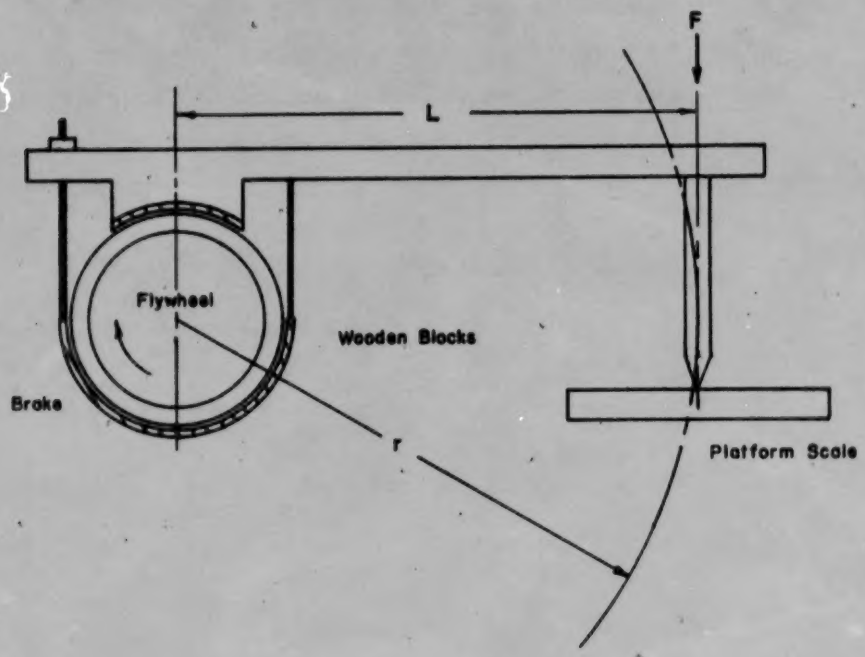
1. Overall thermal efficiency.
2. The influence of fuel-air ratio on engine performance.
3. The effect of varying compression ratio on thermal efficiency.
4. The effects of various fuels on horsepower output.
5. The effect of ignition timing on engine performance.

Once the proper procedure for a specific test has been established, an understanding of basic thermodynamic principles is needed to analyze the data and calculate the results.

The thermal efficiency of a heat engine is the ratio of the measured output to the measured input. A dynamometer of some nature is used to determine the shaft output (brake horsepower) of the engine on test. Basically, there are two types of dynamometers; one classified as an absorption type, and the other referred to as a transmission type. We will restrict ourselves to the former.

The absorption dynamometer may be further categorized as to its method of construction; mechanical, hydraulic or electrical. Regardless of its type of construction, the absorption dynamometer dissipates the power being measured by converting it to heat, either electrically or by friction. The prony brake is typical of the mechanical dynamometers and the following is an analysis of the

principles involved in determining the output of the engine. As shown in the following diagram, the breaking action is due to mechanical friction between the brake shoe and the rotating drum, while the power generated by the engine is dissipated in friction.



The brake horsepower output from the driving engine can be measured by determining the torque (FL) to hold the brake arm in equilibrium.

$$\text{Brake Horsepower (B.H.P.)} = \frac{2\pi\ L\ F\ N}{33{,}000} \qquad (1)$$

L = length of torque arm (ft)

F = net brake load (lb_f)

N = Revolutions per minute of fly wheel

33,000 constant with units of ft-lbf/hp-min

It should be noted that "F" is the net brake load. If the brake arm and brake holder are not balanced initially, it will be necessary to determine the no-load reading of the scale. (Tare weight). When the system is in operation, the scale reading will indicate the gross load, and it will be necessary to subtract the tare weight from the gross weight to obtain the net weight.

The fuel flow rate can be determined by several techniques. The most direct method would be to allow the engine to run at a given load for a set period of time and weigh the fuel at the beginning and end of the test. The difference between the initial and final weights would be the number of pounds of fuel consumed by the engine for this particular test run. The same results could be obtained if a predetermined weight of fuel was used by the engine over a measured period of time.

$$\text{Specific Brake Fuel Rate} = \frac{\text{lb fuel}}{\text{hr}} \times \frac{1}{\text{Brake H.P.}}$$

$$\text{S.B.F.R.} = \frac{\text{lb fuel}}{\text{Brake H.P.} - \text{hr}} \qquad (2)$$

The total energy input per brake horsepower is the product of the specific brake fuel rate and the higher heating valve of the fuel in Btu's per pound of fuel. To determine heating value of distillate fuel with an A.P.I. reading above 18, the following emperical equation may be used.

$$\text{Heating Value (Btu/lb fuel)} = 18{,}250 + 40(\text{A.P.I. reading} - 10) \qquad (3)$$

The thermal efficiency is the ratio of the total output in brake horsepower or kilowatts to the total energy input in Btu's per hour.

$$\text{Thermal Efficiency} = \frac{\text{Output (B.H.P.)}}{\text{Input (Btu/hr)}} \qquad (4)$$

It now becomes necessary to convert the numerator of equation 4 to the same set of units that appear in the denominator. The constant 2,544 Btu/B.H.P. - hr is used for this purpose, and the efficiency equation becomes:

$$\text{Thermal Efficiency} = \frac{\text{B.H.P. x 2544 Btu/B.H.P. - hr}}{W_f \text{ (lb fuel/hour) H.V. (Btu/lb fuel)}} \qquad (5)$$

It should be noted that if both numerator and denominator were to be divided by the brake horsepower the equation for thermal efficiency will be reduced to the following:

$$\text{Thermal Efficiency} = \frac{\text{2544 Btu/B.H.P. - hr}}{\dfrac{W_f \text{ (lb fuel/hour) H.V. (Btu/lb fuel)}}{\text{B.H.P.}}} \qquad (6)$$

where $W_f/\text{B.H.P.}$ = Specific brake fuel rate (S.B.F.R.)

therefore:

$$\text{Thermal Efficiency} = \frac{\text{2544 (Btu/B.H.P.- hr)}}{\text{S.B.F.R. (lb fuel/B.H.P.-hr) H.V.(Btu/lb fuel)}} \qquad (7)$$

The power developed by an engine depends upon the rate at which chemical energy, in the form of fuel, is supplied to the engine and the efficiency with which this energy is converted into useful work.

The combustion process converts the chemical energy of the fuel into internal energy within the cylinder. Oxygen, from the air, must be supplied in sufficient quantities to insure that the combustion process is carried out efficiently. There is an optimum relation-

ship between the pounds of air supplied to support combustion and the number of pounds of fuel used. This relationship is known as the fuel to air ratio and it will vary dependent on the chemistry of the fuel.

Since the amount of fuel that can be introduced into the cylinder on the intake stroke depends on the amount of air present. The air capacity of the engine, in the limit, governs the output.

In a four-stroke cycle engine the ideal air capacity can be determined by the following equation:

$$\text{Air capacity } (ft^3/min) = V_d \times N \times C$$

where V_d (volumetric displacement) $= \frac{.7854\ d^2 L}{1728}$ (ft^3)

d =	engine bore	inches
L =	engine stroke	inches
N =	number of suction strokes per minute	$\frac{(R.P.M.)}{2}$
C =	number of cylinders	

The mass of air based on the air capacity will be equal to the air capacity in ft^3/min times the density of intake air.

$$\dot{m}_{air}\ (lb/min) = V_d(ft^3/min)\ \rho_{air}(lb/ft^3)$$

This is not the actual amount of air ingested by the engine and the volumetric officiency (η_v) expresses the ratio of the actual air capacity to the ideal air capacity.

$$\eta_v = \frac{\text{Actual mass (lb/min)}}{\text{Ideal mass (lb/min)}}$$

where The actual mass is calculated with data obtained from some flow measuring device such as a flow nozzle or rotameter.

The actual amount of air entering the engine is dependent upon the manifold design, valve design lift and timing, engine speed and the engine's condition. The volumetric efficiency, as defined by the above equation, can be made greater than unity by the use of a supercharger.

In the previous discussions, we have found that the thermal efficiency was equal to output over input. It can be shown that the efficiency calculated on the basis of an air-cycle (Ideal) analysis depends only upon compression ratio and the value of the specific heat ratio "K" for the compression process. The efficiency equation becomes:

$$\eta = 1 - \frac{1}{(V_1/V_2)^{K-1}}$$

where V_1/V_2 = Compression Ratio

By definition, compression ratio is the ratio of the cylinder volume when the piston is at the botton of its stroke (V_1) to the volume when the piston is at the top of its stroke (V_2). In the above equation, when the compression ratio increases the efficiency increases. For an actual engine operating at a constant air-fuel ratio and rpm the efficiency will increase but will be much lower than the air cycle efficiencies due to various losses. Increasing compression ratio causes the peak pressure in the engine to also increase. This increase in pressure creates what is considered to be a major factor in limiting the compression ratio of an engine. This factor is detonation. During the combustion process, the flame travels across the combustion chamber increasing the pressure and temperature of the remaining portion of the unburned fuel-air mixture. When certain conditions of pressure, temperature and density of this unburned portion have been met, it may auto-ignite

and burn almost instantaneously. This will release energy at a much greater rate than during normal combustion, and it is this rapid release of energy which causes an audible knock in the engine. This condition is known as detonation.

There are many ways to increase the efficiency of a given engine. If an increase in compression ratio alone is used, care must be taken to protect the engine from excessive pressures and temperatures, and increased frictional effects. A high anti-knock quality fuel must also be used.

Advancing the spark from the optimum or normal setting for a given speed causes ignition to take place earlier on the compression stroke. This early ignition requires the piston to do more work on the compression stroke thereby lowering the total work output of the engine. Retarding the spark brings the point of ignition closer to the dead center position creating the same problem as mentioned earlier when our peak pressures were occuring while the piston was on its power stroke.

In order to obtain maximum performance at different speeds and loads, the point at which the spark occurs must be adjustable. This is accomplished automatically by means of a vacuum advance and a centrifugal advance on the engine's distributor. The vacuum advance is dependent upon the vacuum created in the intake manifold, while the centrifugal advance depends on engine speed. On some small engines the spark may be adjusted manually.

In the ideal fuel-air cycle corresponding to the actual spark ignition engine, ignition is assumed to take place at top dead center with instantaneous burning of the mixture (burning at constant volume). In an actual engine, only that small portion of the mixture nearest the spark plug is ignited and the flame travels through the rest of the mixture until it is completely burned. During this combustion period, the crankshaft has turned a certain number of degrees. If the spark occurs at top dead center, the peak pressure would not be reached until the piston had desended a distance in the cylinder on its power stroke. Since the cylinder

volume is greater than the clearance volume at this point, the peak pressure is less than it would have been at top dead center. This drop in pressure lowers the work produced in the cylinder.

There are two terms commonly used in spark ignition engines, namely; advanced and retarded ignition timing. There is no one point which gives the proper firing of the spark plug with respect to piston position. The optimum setting is dependent on engine speed.

STARTING PROCEDURES FOR MEGATECH MARK III - ENGINE

Operating procedures for the Megatech Mark III Engine and the Electric Dynamometer/ Generator

Basic procedures to be learned before starting.

1. Fuel System and Engine Components

   The engine has two separate fuel tanks which can be used for two different fuels or one fuel and water injection. Water should not be used until after the engine has been started and run for at least 10 minutes. The water filled tank line should always be connected to the carburetor needle valve farthest from the cylinder head.

2. Variable Compression Ratio

   Two compression ratios 3 to 1 and 4 to 1 may be obtained by the proper size piston cap.

3. Since the engine is connected to the DG-1 Dynamometer, it must be connected to an auxiliary air supply with a minimum capacity of 2 CFM at 20 psi. Auxiliary air must be "on" during engine loading otherwise the piston rings may be damaged. If the engine is not connected to any loading device, this auxiliary air is not necessary.

How to start the engine.

Warning: Do not under any circumstances, operate without the plastic shield around the cylinder in place. A glass cylinder which is chipped, cracked or visibly damaged should be replaced.

Clamp or Bolt engine to a sturdy platform or heavy bench.

1. Connect a (6 - 12) volt D.C., (10) ampere (minimum), power source to the terminal post at the rear of the dynamometer.

2. Adjust ignition timing lever 3/4ths of the way, up to full advance position.

3. Turn on air supply and regulate until the gage on dynamometer reads 20 psi.

4. Fill fuel tanks, open throttle fully and turn the ignition switch on.

5. With field reversing switch at (+), turn load rheostat fully clockwise and flip the generator mode switch to the start position.

6. When engine starts cranking, back off load rheostat slightly until peak cranking speed is reached. Open fuel line needle valve slowly, while holding a finger momentarily over the carburetor inlet tube to choke the engine, allowing fuel to come from the tank.

7. As soon as engine starts to run, back off load rheostat completely and set mode switch to generator off position (center). Adjust needle valve for smoothest operation. Check the combustion color so that a slightly rich air-fuel mixture is being used. Blue is lean and orange is the fuel rich color.

8. When the desired rpm has been reached and the engine is running smoothly, the load may be applied by setting switch to load position and gradually increasing the load rheostat.

9. Do not operate the engine at full throttle unless it is loaded.

10. Periodically check tightness of head bolts during operation of the engine. 15 inch-pounds is sufficient with a valve clearance of .017 inches.

11. When stopping the engine, the cooling air must be shut off immediately to prevent damage to the glass cylinder.

Note: 1. This engine may be started when it is not connected to a loading device by means of a starting rope. Wind the starting rope tightly around the starting pulley in a

counterclockwise direction.

2. On engines equipped with a fuel flow meter, a means of adding fuel to the tank will greatly facilitate a test. Otherwise, both tanks would have to be used with some method of switching from one to another.

REFERENCES

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Gill, P. W., Smith Jr., J. H., Ziurys, E. J., "Internal Combustion Engines", United States Navel Institute, Fourth edition, 1959.

Taylor, C. F., Taylor, E. S., "The Internal Combustion Engine", International Textbook Co., Scranton, Pennsylvania, Second edition, 1967.

ENGINE EFFICIENCY EXPERIMENT

OBJECT

To determine the optimum horsepower of a single cylinder spark ignition engine by calculating its thermal efficiency and specific fuel consumption.

APPARATUS

Megatech Mark III engine connected to a model DG-1 electric Dynamometer/generator.

Twelve volt D.C. power source

Fuel flow meter

PROCEDURE

a. Thoroughly understand the starting and operating procedures.

b. Fill fuel tanks with alcohol.

c. Start engine and allow a 3 - 4 minute warm-up period.

d. With throttle fully opened, adjust the load to give the desired RPM. Decrease and increase the fuel flow and note the RPM change. Back off from the too rich mixture until smooth running conditions are met and readjust load to give required RPM. Take readings of torque, manifold pressure, cylinder pressure and fuel rate.

e. Repeat step d, increasing or decreasing the speed in increments designated by the instructor. At each load adjust fuel needle valve to optimum conditions.

f. Periodically check head bolt tightness.

EVALUATION OF RESULTS

a. Plot on a single sheet of graph paper, engine thermal efficiency, specific fuel consumption and HP as the ordinate versus RPM as the abscissa.

b. Analyze the results and explain any peculiarities which may exist.

Data Sheet

RPM	Torque (in-lb)	Manifold Pressure (psig)	Cylinder Pressure (psig)	Fuel Flow (mm)

Type of Fuel _____

Heating Value of Fuel _____ Btu/lb

Results Sheet

RPM	Engine Output		Fuel Flow		Input Btu/min	Specific Fuel Consumption lb/hp-hr	Thermal Efficiency (%)
	Horsepower	Btu/min	grams/min *	lb/min			

* From Appropriate Curve

IGNITION TIMING EXPERIMENT

OBJECT

To compare the effect of timing change on the output of a spark ignition engine.

APPARATUS

Megatech Mark III Engine connected to a Model DG-1 Electric Dynamometer/Generator

Twelve volt D.C. Power Source

Timing light

PROCEDURE

a. Thoroughly understand the starting and operating procedures.

b. Fill fuel tanks with the desired fuel.

c. Start engine and set the ignition timing at zero degrees.

d. With the throttle fully opened, adjust the load to give the desired RPM. Decrease and increase the fuel and note the RPM change. Back off from the too rich mixture setting until smooth running conditions are met and readjust load to give required RPM. Take readings of torque, manifold pressure and cylinder pressure.

e. Repeat step d, increasing or decreasing the speed in increments designated by the instructor. At each load, adjust the fuel needle valve so that optimum conditions will be met.

f. Repeat steps d and e with the ignition set at two advanced settings and at two retarded settings.

g. Periodically, check head bolt tightness.

EVALUATION OF RESULTS

a. Plot at each ignition setting a curve using RPM as the abscissa versus HP as the ordinate.

b. Plot at three different speeds, a curve with ignition settings as the abscissa versus HP as the ordinate.

c. Determine from your plots the speed and ignition setting at which the engine should be run.

Data Sheet

Ignition Setting	RPM	Torque (in-lb)	Manifold Pressure (psig)	Cylinder Pressure (psig)
20° Advanced				
10° Advanced				
0°				

Data Sheet

Ignition Setting	RPM	Torque (in - lb)	Manifold Pressure (psig)	Cylinder Pressure (psig)
10° Retard				
20° Retard				

Results Sheet

RPM	Horsepower				
	+20°	+10°	0°	-10°	-20°

AIR - FUEL RATIO EXPERIMENT

OBJECT

To determine what influence the air-fuel ratio has on engine performance.

APPARATUS

Megatech Mark III Engine connected to a Model DG-1 Electric Dynomometer/Generator

Twelve volt D.C. power supply

Fuel flow meter

Air flow meter

PROCEDURE

a. Thoroughly understand the starting and operating procedures.

b. Fill fuel tank with alcohol and start engine.

c. After a 3 to 4 minute warm-up period, adjust the load to give the desired RPM with the throttle fully opened. Decrease and increase the fuel and observe the RPM change. Back off from the too rich mixture setting until smooth running conditions are met and readjust the load to obtain the desired RPM. Take readings of torque manifold pressure, cylinder pressure, fuel rate and air rate.

d. Repeat step c, for three other speeds at full throttle. At each load, adjust fuel needle valve to optimum conditions.

e. Repeat steps c and d with the throttle set at the 3/4, 1/2 and 1/4 open position.

f. Periodically, check head bolt tightness.

EVALUATION OF RESULTS

a. Plot for each throttle setting, a graph using RPM as the abscissa versus HP as the ordinate. Indicate on each curve the throttle setting.

b. Plot for each throttle setting, a graph using RPM as the abscissa versus air consumption (ft^3/hr) and specific fuel consumption as the ordinate. Indicate on each curve the throttle setting.

Data Sheet

Throttle Setting	RPM	Torque in-lb	Manifold Pressure psig	Cylinder Pressure psig	Fuel Rate mm/min	Air Rate cu. ft./hr.
Fully Open						
3/4 Open						
1/2 Open						
1/4 Open						

Barometer Pressure _____ Room Temperature _____

Results Sheet

Throttle Settings	RPM	Horsepower	Fuel Rate lbs/hr	Air Rate cu.ft./hr	Volumetric Efficiency
Fully Open					
3/4 Open					
1/2 Open					
1/4 Open					

ENGINE PERFORMANCE USING DIFFERENT FUELS EXPERIMENT

OBJECT

To determine which fuel when used in a spark ignition engine will give optimum output.

APPARATUS

Megatech Mark III Engine connected to a Model DG-1 Electric Dynamometer/Generator

12 volt D.C. Power Supply

Three fuels - alcohol, gasoline and propane

Fuel Flow Meter

Weighing scales

PROCEDURE

a. Thoroughly understand the starting and operating procedures.

b. Fill fuel tank with the desired liquid fuel and start engine. If propane is used, the cylinder must be placed on a set of weighing scales.

c. After a 3 to 4 minute warm-up period, adjust the load to give the desired RPM with the throttle fully opened. Decrease and increase the fuel and observe the RPM change. Back off from the too rich mixture setting until smooth running conditions are met and readjust load to obtain the required RPM. Take readings of torque, manifold pressure, cylinder pressure and fuel rate.

d. Repeat step c, increasing or decreasing the speed in increments designated by the instructor. At each load, adjust fuel needle valve to optimum conditions.

e. Periodically, check head bolt tightness.

EVALUATION OF RESULTS

a. On the same graph, plot HP versus RPM for each fuel. Use RPM as the abscissa and HP as the ordinate.

b. Compare the fuels used, discussing the advantages and disadvantages of using each.

Data Sheet

Fuel	RPM	Torque (in-lb)	Manifold Pressure psig	Cylinder Pressure psig	Fuel Rate mm/min
Alcohol					
Gasoline					

Fuel	RPM	Torque (in-lb)	Manifold Pressure psig	Cylinder Pressure psig	Fuel Weight lbs	Time sec.
Propane						

Results Sheet

RPM	Alcohol		Gasoline		Propane	
	Horsepower	Fuel Rate lbs/min	Horsepower	Fuel Rate lbs/min	Horsepower	Fuel Rate lbs/min

COMPRESSION RATIO EXPERIMENT

OBJECT

To determine what effect the change in compression ratio has on engine performance.

APPARATUS

Megatech Mark III Engine connected to a DG-1 Electric Dynamometer/ Generator

12 Volt D.C. Power Source

4 Piston caps for 3 to 1 and 4 to 1 compression ratios

Fuel Flow Meter

PROCEDURE

a. Thoroughly understand the starting and operating procedures.

b. With the 3 to 1 piston cap in place, fill fuel tank with the desired fuel and start engine.

c. After a 3 to 4 minute warm-up period, adjust the load to give the desired RPM with the throttle fully opened. Decrease and increase the fuel and observe the RPM change. Back off from the too rich mixture setting until smooth running conditions are met and readjust the load to obtain the desired RPM. Take readings of torque, manifold pressure, cylinder pressure and fuel rate.

d. Repeat step c for other speeds designated by the instructor. At each load, adjust fuel needle valve to optimum conditions.

e. Remove cylinder head, cylinder shield and cylinder from the engine. Replace the 3 to 1 piston cap with the 4 to 1 cap making sure the cotter pin is in place. Using new cylinder gaskets, install the cylinder, shield and head onto the engine. Torque head bolts to 15 inch-pounds and set valves to .017 inches.

f. Repeat steps c and d for this new compression ratio.

g. Check tightness of head bolts periodically throughout test.

EVALUATION OF RESULTS

a. Plot for each compression ratio, a graph with RPM as the abscissa versus HP, thermal efficiency and specific fuel consumption as the ordinate.

Data Sheet

Comp. Ratio	RPM	Torque in-lb	Manifold Pressure psig	Cylinder Pressure psig	Fuel Rate mm/min
3 to 1					
4 to 1					

Fuel Used _____ Heating Value _____

Results Sheet

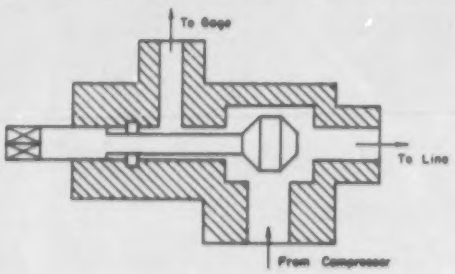
RPM	3 to 1 Compression Ratio			4 to 1 Compression Ratio		
	Horsepower	Thermal Efficiency	sfc lbs/hp-hr	Horsepower	Thermal Efficiency	sfc lbs/hp-hr

The operation of a vapor compression systems requires that certain mechanical skills must be acquired by the technician. These skills involve not only an operational knowledge of the cycle, but the ability to use those tools and devices that are peculiar to the field.

The service manifold is one of the most versatile instruments that the technician has available to determine the operating conditions within a refrigeration system.

The manifold consists of two hand operated valves, two gauges and three line connections. With the proper adjustment of valves and line connections, it is possible to ascertain pressures at significant points with the system.

To facilitate service, compressors are normally equipped with special valves on the suction and discharge lines. These valves are known as "back seating" valves and can be set in three positions. (see following diagram).

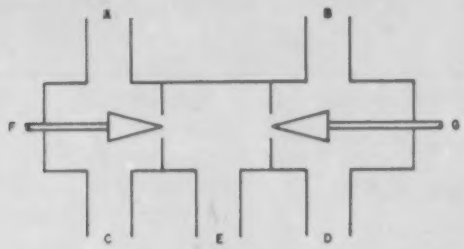


Service Valve

1. In the back seat position the gauge port is closed, but valve is open and system may operate.
2. In front seat position, line connection to compressor is shut off, but gauge port is open to compressor. System should not be operated under these conditions.
3. In mid position both line and gauge ports are open to compressor. System may operate under these conditions.

A brief description of the physical arrangement of the service manifold and some of the connections that should be made for various mechanical operations follows:

I. Physical Arrangement



Manifold

A. Compound Gauge
B. High Pressure Gauge
C. Line connection to suction service valve
D. Line connection to discharge service valve
E. Service opening
F. Needle valve
G. Needle valve

II. Manifold connections to perform following operations:

A. Low Side Charging
   1. Connect C and D respectively to low and high side service valves respectively.
   2. Close valve G.
   3. Connect service drum with charging line at E.
   4. Open valve F.

B. System Evacuation
   1. Connect C and D to low and high sides respectively.
   2. Connect E to vacuum pump.
   3. Open valves F and G.

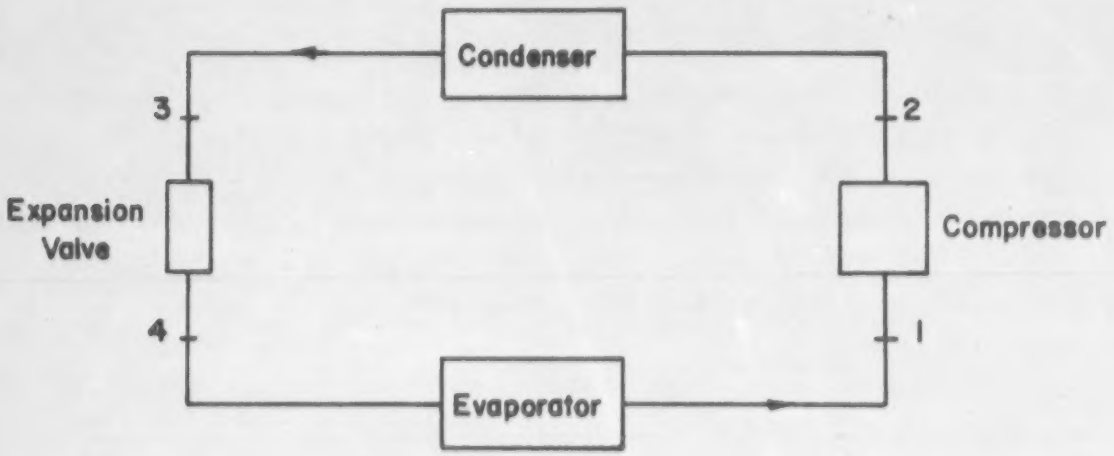
C. Gauge Reading

1. Connect C and D to low and high sides respectively.

2. Close valves F and G.

In order to properly diagnose the mechanical problems typical to vapor compression systems, a knowledge of basic thermodynamics is essential.

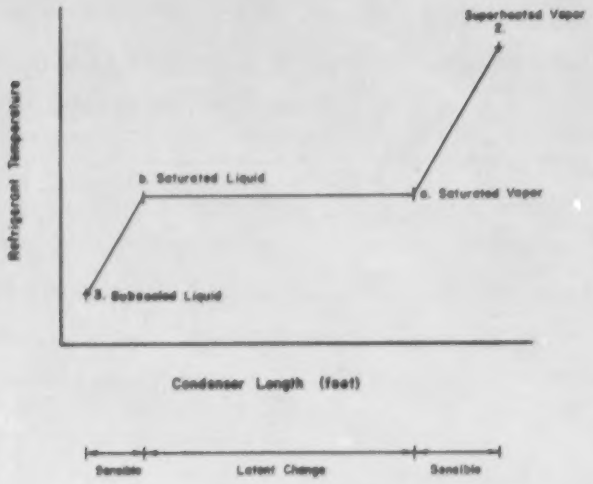
All vapor compression systems use a refrigerant that undergoes various processes as a cycle of events is completed. The working substance (refrigerant) is alternately compressed, condensed, expanded and vaporized during the cycle of operation, and the mechanical equipment needed for the performance of these processes is indicated in the following schematic:



The function of the compressor is to remove, at a constant rate, the heat laden refrigerant from evaporator and increase the pressure of the working substance so that its saturation temperature will be greater than the temperature of the cooling medium used in the condenser.

The vapor leaves the compressor in a superheated state and enters the condenser. A heat rejection process occurs and the refrigerant experiences a sensible change in temperature to saturation, followed by a phase change in which condensation takes place at constant

pressure. At this point, the refrigerant, now a saturated liquid, is further cooled and leaves the condenser subcooled.



The subcooled liquid, at condenser operating pressure, is fed to the expansion valve and a metered amount of fluid is expanded to evaporator pressure. As the fluid expands from condenser pressure to that maintained in the evaporator, the refrigerant "flashes" from a subcooled liquid to a wet vapor and assumes the saturation temperature corresponding to evaporator pressure. This expansion process is referred to as a throttling process and is carried out at constant enthalpy.

The wet vapor at low temperature is now capable of absorbing heat from the surroundings and the refrigerant vaporizes at constant pressure. Ideally, the working substance would leave evaporator as a dry vapor, but to insure that no liquid enters the compressor the vapor leaving the evaporator is superheated. The refrigerant now enters the compressor and the cycle of operations begins anew.

The following sequence of experiments is designed to acquaint the students with problems peculiar to refrigeration systems.

C. Gauge Reading

1. Connect C and D to low and high sides respectively.

2. Close valves F and G.

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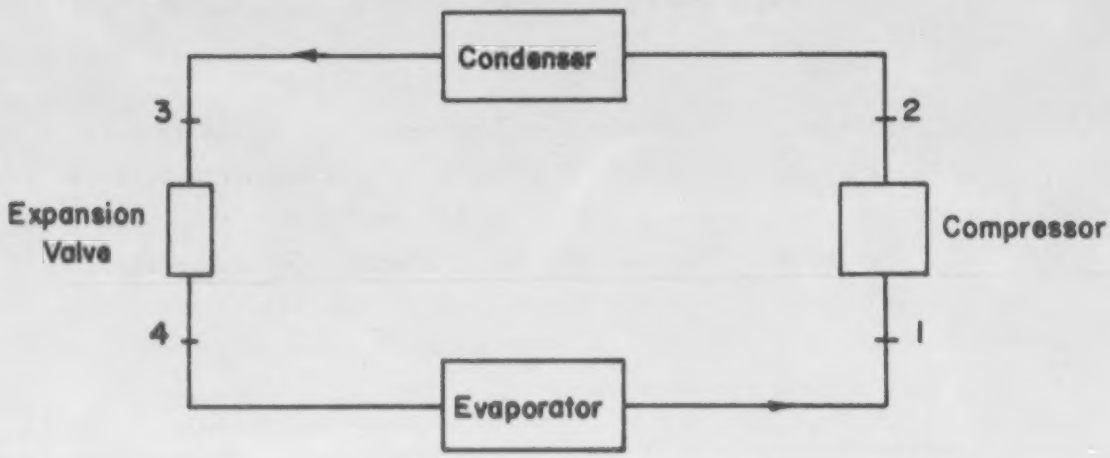

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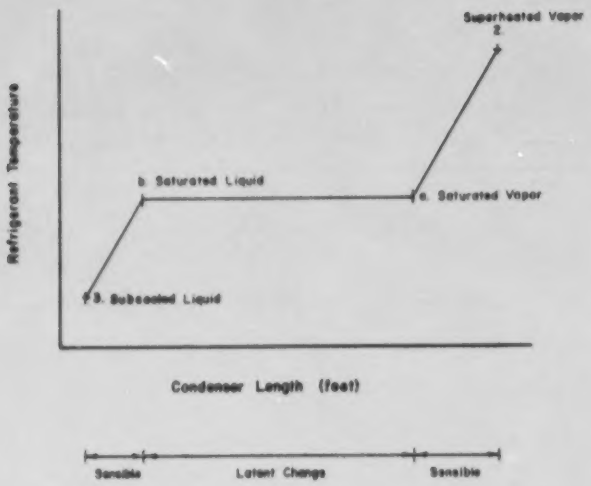

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The following sequence of experiments is designed to acquaint the students with problems peculiar to refrigeration systems.

VAPOR COMPRESSION SYSTEM ANALYSIS

OBJECT

a. To identify the physical components that make up a typical vapor compression system.

b. To determine the properties of the refrigerant at significant points in the cycle.

APPARATUS

Vapor compression system operating on Refrigerant-12

Service manifold

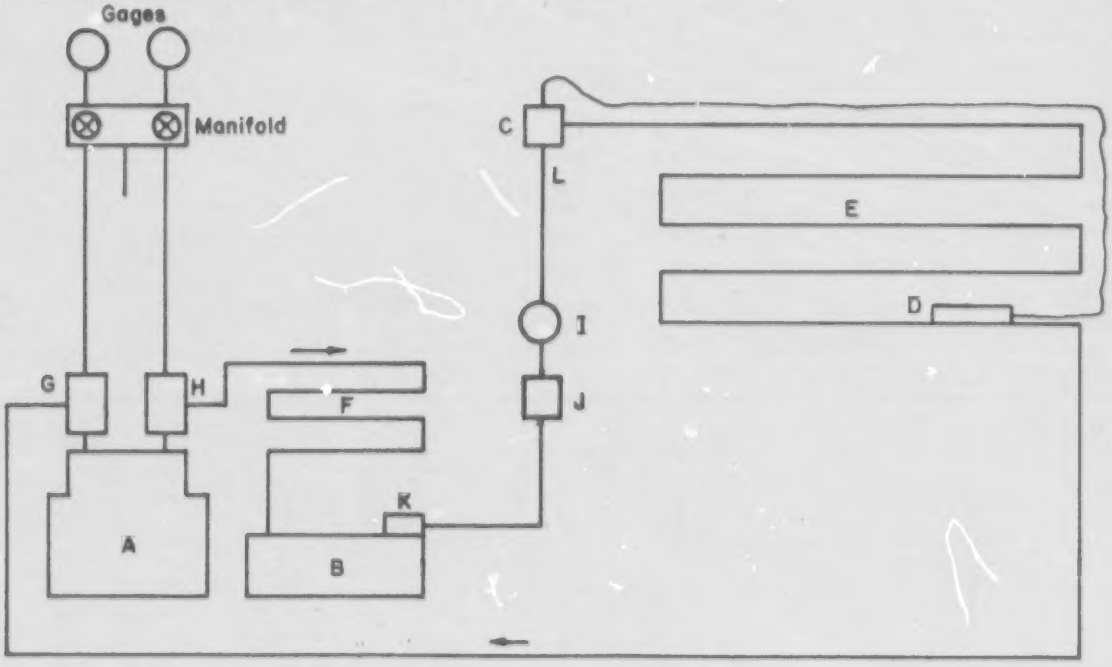
Service wrenches

Thermistor

PROCEDURE

a. Visually inspect system, identify all components and record on appropriate data sheet.

b. Back seat suction and discharge service valves and connect manifold as indicated in the following diagram.



c. Crack manifold connection at high side and purge any air that may be in line by manipulation of high side service valve. Do same for low side. Close off service valves and tighten manifold connections.

d. Attach leads from thermister so that the temperatures may be read at points G, H, L and D.

e. Have instructor check out system to make sure all connections are correct and that the unit is properly instrumented.

f. Start up unit and adjust service valves until system pressure is registered. Excessive vibration of needle can be controlled by manipulation of service valve.

g. After system stabilizes, record pressures and temperatures at 10 minute intervals for a test run of 30 minutes.

h. Back seat high side service valve and open both manifold valves. When manifold pressure has equalized back seat suction service valve and remove gauges.

i. Shut down unit.

EVALUATION OF RESULTS

a. Using the data recorded, determine the enthalpy of the refrigerant at following points and record.

   1. Compressor suction
   2. Compressor discharge
   3. Entrance to expansion valve
   4. Evaporator exit

b. Plot these points on the p-h diagram provided.

Data and Results Sheet

Location	Name of Component
A	
B	
C	
D	
E	
F	
G	
H	
I	
J	
K	

Location	Pressure	Temperature	Enthalpy
Compressor Suction			
Compressor Discharge			
Expansion Valve Inlet			
Evaporator Exit			

MECHANICAL SERVICING EXPERIMENTS

OBJECT

a. Develop techniques which will enable the technician to perform the following mechanical operations.

1. System evacuation
2. System charging
3. Pumping down a system
4. Setting pressure controls

APPARATUS

Vapor Compression system operating on Refrigerant-12

Service manifold

Service wrenches

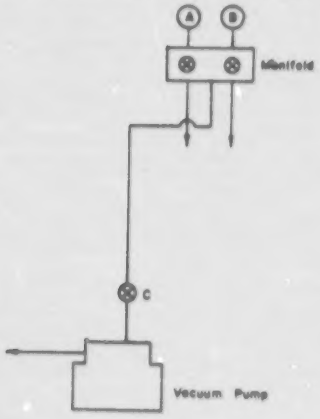
Vacuum pump with micron gauge

Service Cylinder

SYSTEM EVACUATION

PROCEDURE

a. Connect vacuum pump to system as shown in following diagram.



A. Compound gauge
B. High pressure gauge
C. Vacuum pump valve connection

b. Open suction and discharge service valves on compressor.

c. Open both high and low side valves on manifold.

d. Crack open line connection at valve C to insure that system is at atmospheric pressure before evacuation.

e. Tighten line connection at C and open valve wide.

f. Start vacuum pump and pull system down to recommended range for dehydration.

g. Shut down pump and note any increase in vacuum gauge reading.

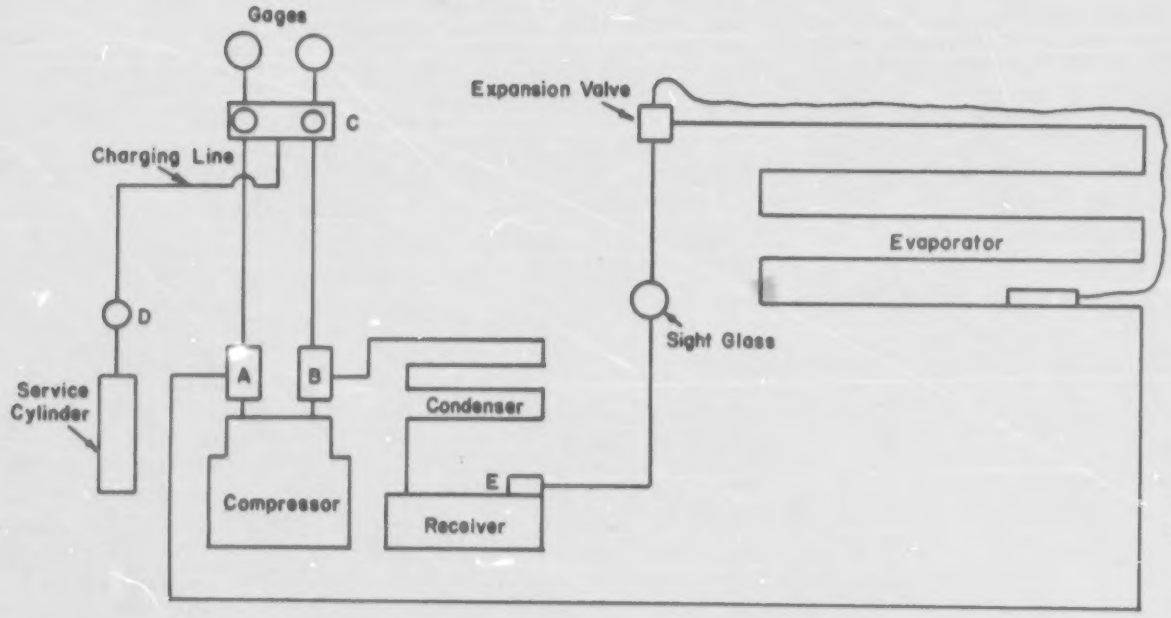
h. If any increase in pressure is registered, it will be necessary to start vacuum pump again and continue pump-down process.

i. Continue evacuation process until pressure stabilizes at desired conditions.

SYSTEM CHARGING

PROCEDURE

a. Connect service cylinder to system as shown in following diagram.



b. Back seat high and low side service valves and connect manifold to system.

c. Connect charging line from manifold to service cylinder at valve D.

d. Crack open valve D and purge air from charging lines.

e. Crack open low side manifold valve and purge air up to service valve A. Do same for high side. Then close off high side valve.

f. With the low side service valve open, allow refrigerant vapor to enter suction side of compressor through valve D.

g. Start up compressor (high side pressure should increase and low side pressure should pull-down) and allow vapor to enter system.

h. When sight glass is clear, i.e., no bubbles, system is assumed to be charged.

i. Close valve D and back seat high side service valve B. Open both manifold valves and allow pressures shown on gauges to equalize.

j. Back seat suction service valve A and remove all connections.

k. Shut down system.

PUMPING DOWN A SYSTEM

PROCEDURE

a. Connect manifold gauges to system as in previous experiments. Make sure valves are closed on manifold.

b. Front seat liquid line service valve.

c. Start unit and note pressures. When suction pressure falls to approximately 1/2 psi, stop compressor.

d. If pressure increases, repeat preceding step until system pressure becomes stabilized at 1/2 to 1 psi.

e. System is now pumped down and the refrigerant has been stored in the receiver.

SETTING PRESSURE CONTROLS

GENERAL INFORMATION

I. A control is nothing more than a switching device that can be used to control the operation of various components within the the refrigeration system or the motor compressor itself. It is important that the technician understands how these devices work and must be capable of setting the controls for differing sets of conditions.

II. Terminology Applicable to Controls

A. Cut-in (point)_ This is the pressure at which control closes switch and makes circuit.

B. Cut-out (point)-This is the pressure at which control opens switch and breaks circuit.

C. Differential - Difference between cut-in and cut-out.

III. Table of Approximate Control Settings using Refrigerant-12

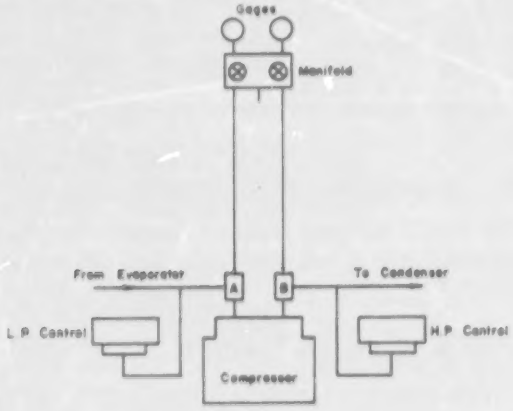
Table of Approximate Control Settings Using Refrigerant 12

Application	Pressure Setting	
	Cut-Out	Cut-In
Walk-In Cooler	13	35
Reach-In Cooler	18	36
Vegetable Case	20	40
Ice Cuber	4	17
Florist Box	28	42

Cut-in point (pressure) will correspond to maximum permissible temperature for application and the cut-out point will vary depending on the design temperature difference.

PROCEDURE

a. The following diagram illustrates arrangement of equipment to be used in setting pressure controls.



b. Using table of pressure settings, set cut-in and differential for a walk-in cooler.

c. Start compressor and allow unit to run. Note low pressure reading on gauges.

d. Throttle receiver valve and allow compressor to pump down.

e. As suction pressure decreases, note the pressure at which compressor stops.

f. Compare gauge reading at this point to recommended cut-out point.

g. If actual cut-out and recommended cut-out point are in close agreement, continue experiment; otherwise open receiver valve and reset cut-out properly.

h. To check cut-in slowly, open receiver valve and notice increase in suction pressure.

i. Read pressure at point when compressor starts.

j. If there is reasonable agreement, the control has been set properly.

k. Repeat steps b through j for other applications as assigned by instructor.

SYSTEM TONNAGE

Once a system has been installed there are several methods that can be used to determine system tonnage. A common technique is to determine the refrigeration effect from an analysis based on the changes in the properties of the air as it passes through the coil.

The underlying principle in this method is simply that the heat removed from the air is equal to the heat absorbed by the refrigerant. The heat removed from the air, as a positive number, can be calculated by the following:

$$Q_{Air} = \dot{m}_{Air}(h_1 - h_2)_{Air} \qquad (1)$$

Q = heat removed from air	Btu/min
$\dot{m}$ = weight of air/unit time	lb/min
h_1 = enthalpy of entering air	Btu/lb
h_2 = enthalpy of leaving air	Btu/lb

Now it becomes a matter of determining the quantity of air flowing per unit of time and the various properties of the air entering and leaving the coil. The wet bulb and dry bulb temperatures can be measured with a sling psychrometer and then using a psychrometric chart all other pertinent properties may be determined.

The continuity equation can be used to calculate the volume rate of flow.

$$\dot{V} = Av \qquad (2)$$

$\dot{V}$ = volume rate of flow	ft^3/min
A = cross sectional area of duct	ft^2
v = velocity of air at duct exit	ft/min

(If a grille is used at duct exit, a grille factor must be applied).

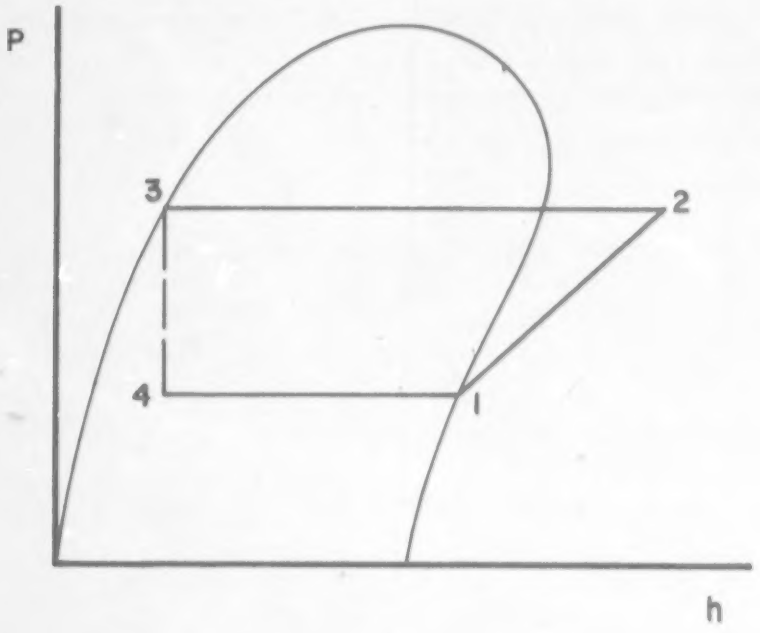
To convert the volume rate of flow to a mass flow rate, use the following equation.

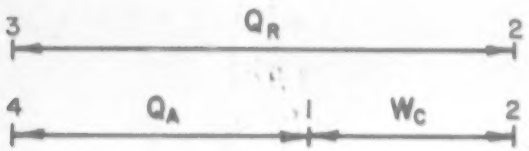
$$\dot{m} = \rho \dot{V} \qquad (3)$$

$\dot{m}$ = mass flow rate	lb/min
ρ = density of leaving air	lb/ft^3
$\dot{V}$ = volume flow rate	ft^3/min

Using Eq-1, the total heat removed from the air can be calculated and dividing this value by 200 Btu/min-ton the tonnage of the system is determined.

Another method that may be used with systems using water cooled condensers, is based on conditions in the condenser. From thermodynamics, it can be shown that the heat rejected to the condenser cooling water is equal to heat absorbed by the refrigerant in the evaporator plus the work of compression. For the ideal cycle, this can be illustrated on a p-h diagram in the following manner.





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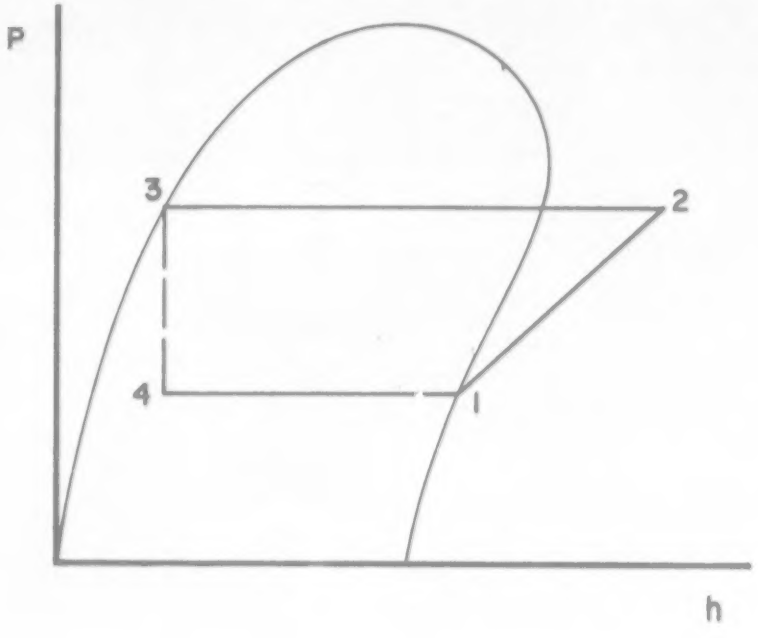

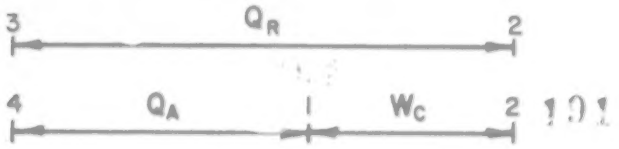

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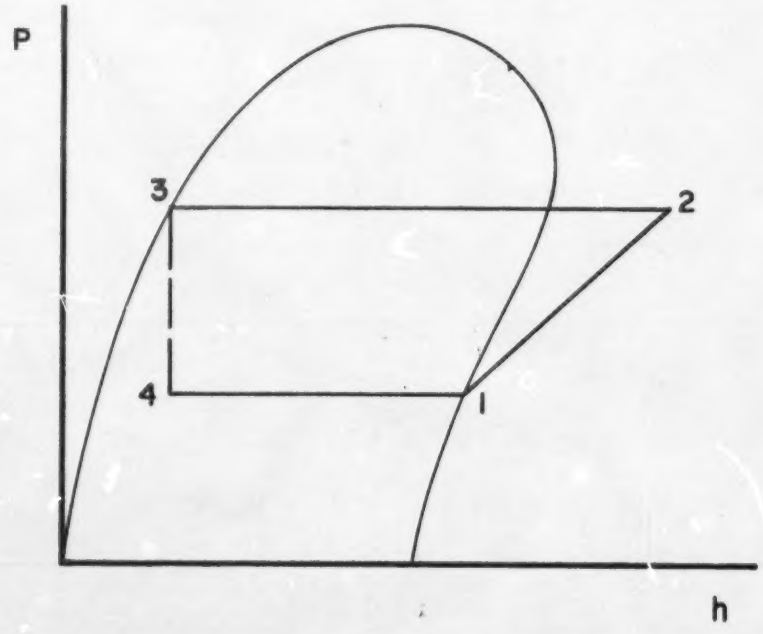
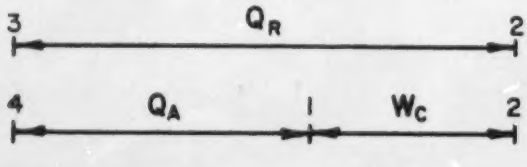




The amount of heat rejected to the condenser will vary as a function of the compressor suction and discharge pressures. Ratios of heat rejection to heat absorbed are available from manufacturer's catalogs and are used to determine this ratio for a variety of operating conditions. Referring to the p-h diagram, it can be seen that:

$$Q_R = Q_A + W_c \tag{4}$$

$\frac{Q_R}{Q_A}$ = heat rejection ratio (H.R.R.)*

$$Q_R = \frac{Q_R}{H.R.R.} + W_c \tag{5}$$

$$Q_R \{ 1 - \frac{1}{H.R.R.} \} = W_c \tag{6}$$

$$Q_A = Q_R - Q_R \{ 1 - \frac{1}{H.R.R.} \} \tag{7}$$

The heat rejected by the refrigerant as it passes through the condenser is equal to the heat gain of the condenser cooling water.

$$Q_{R\ (refrig.)} = Q_{gained\ (cooling\ water)}$$

$$Q_{gained} = \dot{m}\, c_p\, (T_{out} - T_{in}) \tag{8}$$

Q = heat gained by cooling water	Btu/min
$\dot{m}$ = mass flow rate	lb/min
c_p = specific heat of water	Btu/lb ^oF
T_{out} = cooling water temperature (outlet)	^oF
T_{in} = cooling water temperature (inlet)	^oF

* Obtained from manufacturer's catalogs.

Using equations 7 and 8, necessary computation can be made to determine the heat absorbed by the refrigerant in Btu/min. This can now be converted to tonnage simply by dividing the result of this calculation by 200 Btu/min-ton.

If catalogs are not available for the heat rejection to heat absorbed ratio, it will be necessary to assume an average value for work of compression. In this case the following equation may be used to determine the tonnage.

$$\text{Tonnage} = \frac{K\, c_p\, \dot{V}\, \rho(\Delta T)}{C} \qquad (9)$$

$K = 0.86$

c_p = specific heat of water — Btu/lb ^{o}F

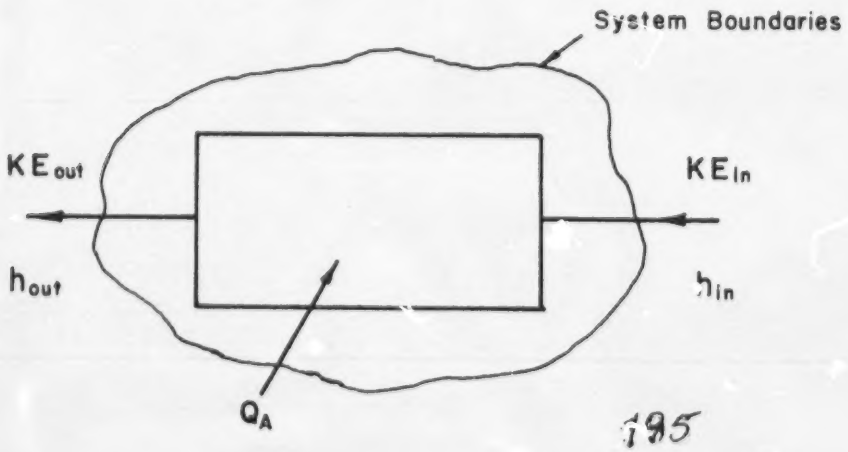
$\dot{V}$ = flow rate of water — ft^3/min

ρ = density of water — lb/ft^3

ΔT = change in temperature of cooling water — ^{o}F

C = conversion constant — 200 Btu/min-ton

The third and final method is the determination of tonnage from the refrigerant side. For this case, consider the refrigerant as the thermodynamic system and complete an energy balance as shown:



Energy In = Energy Out

Energy In = Energy Out

$$KE_{in} + \dot{m} h_{in} + Q = KE_{out} + \dot{m} h_{out}$$

(Neglecting the effect of changes in kinetic energy and solving for Q.)

$$Q = \dot{m} (h_{out} - h_{in}) \quad \text{Btu/min} \quad (10)$$

Q = heat added to refrigerant	Btu/min
$\dot{m}$ = mass flow rate of refrigerant	lb/min
h_{in} = enthalpy of vapor entering coil	Btu/lb
h_{out} = enthalpy of vapor leaving coil	Btu/lb

The enthalpy of the wet vapor entering the coil is evaluated by measuring the temperature and pressure of the subcooled liquid at inlet to expansion valve. The developed pressure (in this case) will not significantly affect the value of the liquid properties. Therefore, the temperature can be used to determine the enthalpy of the liquid and only introduce a slight error in the calculations. The process across the valve is a throttling process and the enthalpy at valve inlet is equal to enthalpy at valve exit (inlet to coil). At the evaporator exit the vapor is superheated, and knowing the temperature and pressure at this point, the enthalpy can be determined. The mass flow rate is measured by placing a calibrated meter in the liquid line.

With this information and using equation 10, the refrigeration in Btu/min can be computed. Dividing this value by 200 Btu/min-ton will result in system tonnage.

REFERENCES

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Althouse, A. D., Turnquist, C. H., and Bracciano, A. F., "Modern Refrigeration and Air Conditioning", The Goodheart-Wilcox Company, Inc., Homewood, Illinois, 1968.

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Tuve, G. L., "Mechanical Engineering Experimentation", McGraw-Hill, New York, 1961.

REFRIGERATION TONNAGE EXPERIMENT

OBJECT

To determine the tonnage delivered by a vapor compression system operating on Refrigerant - 12 by three methods.

APPARATUS

Coil and condensing unit
Manifold gauges
Service wrenches
Thermistor
Sling psychrometer
Biram anemometer

PROCEDURE

a. Instrument unit to measure temperatures and pressures as indicated on data sheets.

b. Start up unit and allow to run for ten minutes to stabilize operating conditions.

    1. Adjust water regulating valve, if necessary, to maintain the head pressure between 115-125 psig.

    2. Suction pressure should be about 40 psig.

c. Run test for period of time as designated by instructor and record data as indicated on data sheet.

d. At conclusion of test, shut down unit and remove portable instrumentation.

e. Calculate the following:

    1. Total heat removed from air Btu/hr.
    2. Sensible heat removed from air Btu/hr.
    3. Latent heat removed from air Btu/hr.

4. Tons of refrigeration from air side tons
5. Tons of refrigeration from water side tons
6. Tons of refrigeration from refrigerant side tons

EVALUATION OF RESULTS

a. Check manufacturers catalog for rated tonnage of unit and compare rating against calculated values.

b. Which method of calculating tonnage most closely approximates the rated capacity?

c. If a significant difference is noted, discuss the possible reasons for deviation.

Data Sheet

Air Side **Water Side**

Time	Temperature (°F)				Duct Velocity fpm	Condenser Water Temp. (°F)	
	Dry Bulb (in)	Wet Bulb (In)	Dry Bulb (Out)	Wet Bulb (Out)		In	Out
0							
mid.							
end							
average							

Anemometer Correction _____ **Duct Area** _____ **sq. ft.** **Meter Reading: Intial** _____

Rotameter Correction _____ **Final** _____

Refrigeration Side

Time	Pressure (psig)			Temperature (°F)		Rotameter Reading gpm
	Comp. Suct.	Comp. Disch.	Evaporator	Exp. Valve (In)	Evaporator (Out)	
0						
mid.						
end						
average						

Results Sheet

Air Side

- Enthalpy Btu/lb
  - Air In _____
  - Air Out _____
- Density of Air Out lb/cu. ft. _____
- Corrected Velocity fpm _____
- Mass-Flow-Rate lb/min. _____
- Moisture Content lbv/lb d.a.
  - Air In _____
  - Air Out _____
- Heat Removed Btu/min
  - Total _____
  - Sensible _____
  - Latent _____
  - Refrigeration Effect tons _____

Water Side

- Mass-Flow-Rate lb/min. _____
- ΔT Across Condenser _____
- Specific Heat of Water Btu/lb-°R _____
- Total Heat Rejected Btu/min. _____
- Compressor Work Btu/min. _____
- Refrigeration Effect tons _____

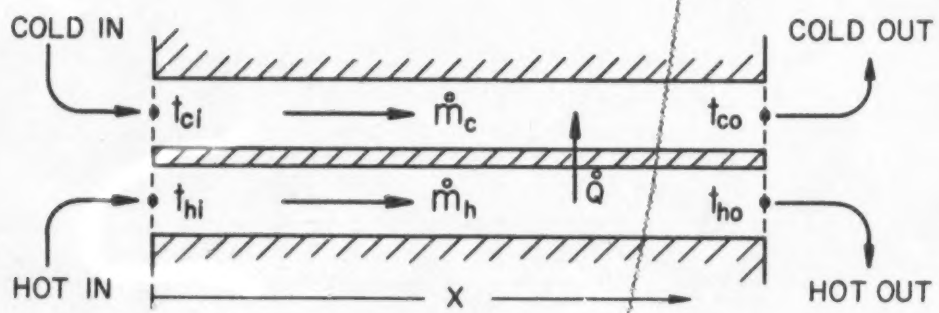
Refrigeration Side

- Enthalpy Btu/lb
  - Evaporator Out _____
  - Evaporator In _____
- Δh Across Evaporator Btu/lb _____
- Mass-Flow-Rate lb/min. _____
- Refrigeration Effect tons _____

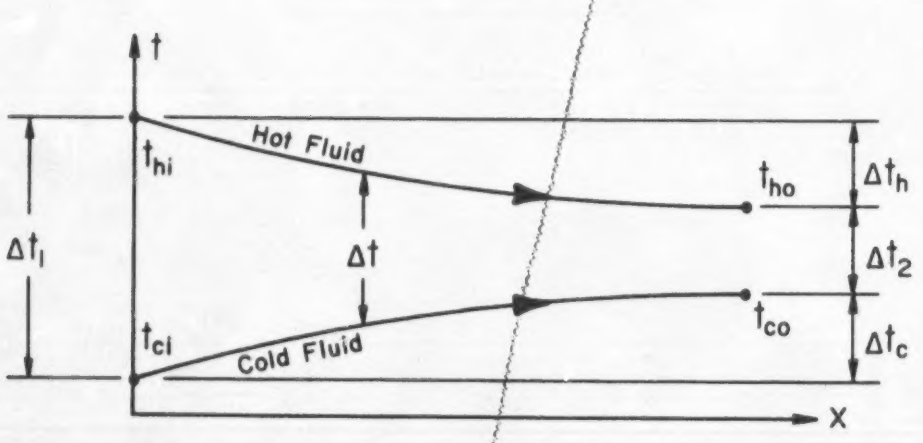
Heat exchangers are devices used to transfer heat from one substance to another - usually from one fluid to another, with or without separation by a wall. Separated fluid heat exchangers may be either of the rotary type, or of the static type.

The static, separated fluids heat exchangers are usually divided into three groups, depending on the flow directions of the fluids relative to each other:

1. Parallel flow
2. Counter flow
3. Cross flow



(a) Flow Diagram



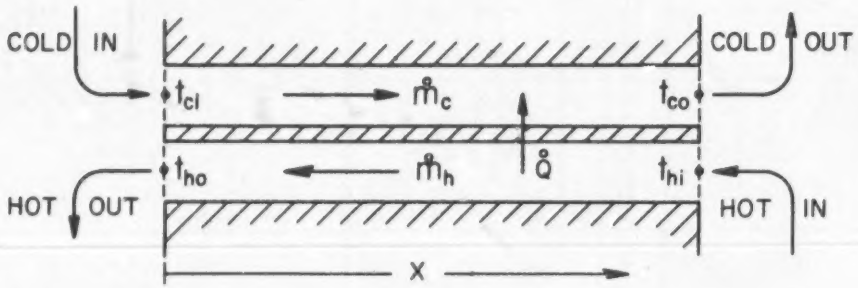
(b) Temperature Distribution

Fig. 1 PARALLEL FLOW HEAT EXCHANGER

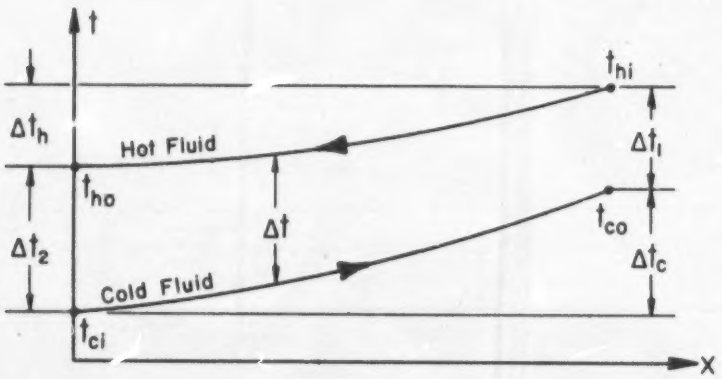
In the foregoing and following diagrams:

$\dot{Q}$	= rate of heat transfer	Btu/hr
t_{ci}	= cold fluid inlet temperature	^{o}F
t_{co}	= cold fluid outlet temperature	^{o}F
t_{hi}	= hot fluid inlet temperature	^{o}F
t_{ho}	= hot fluid outlet temperature	^{o}F

In the parallel flow heat exchanger, the two fluids flow in the same direction. The hottest part of the hot fluid exchanges heat with the coldest part of the cold fluid. There is a large temperature difference at the inlet, and a gradually diminishing difference towards the outlet.



(a) Flow Diagram



(b) Temperature Distribution

Fig. 2 COUNTER FLOW HEAT EXCHANGER

In the counter flow heat exchanger, the two fluids flow in opposite directions. The hottest part of the hot fluid exchanges heat with the hottest part of the cold fluid, and the temperature difference is more nearly constant over the length of the exchanger. (See Figure 2).

In a cross flow heat exchanger, the two fluids flow at right angles to each other. The most common example of this type of heat exchanger is the automobile radiator. The temperature distribution in the cross flow heat exchanger is considerably more complex.

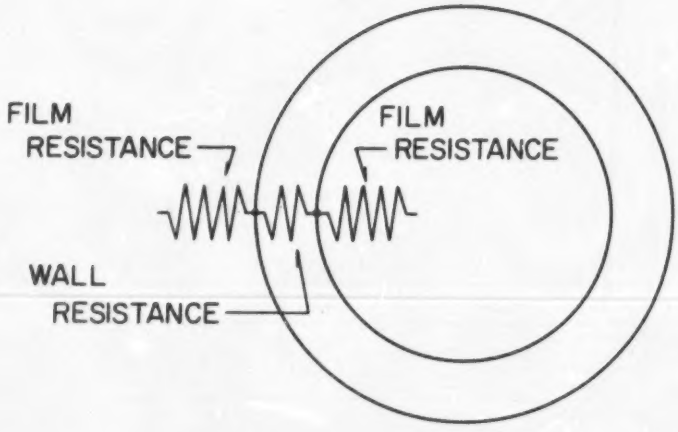


Fig. 3 THERMAL RESISTANCES IN CONVECTION

If we consider the internal wall of the heat exchanger through which heat is transferred from a hot fluid to a cold fluid, we recognize, at least, three resistances in series:

1. The resistance between the wall and the hot fluid (film resistance).

2. The resistance of the wall.

3. The resistance between the wall and the cold fluid (film resistance).

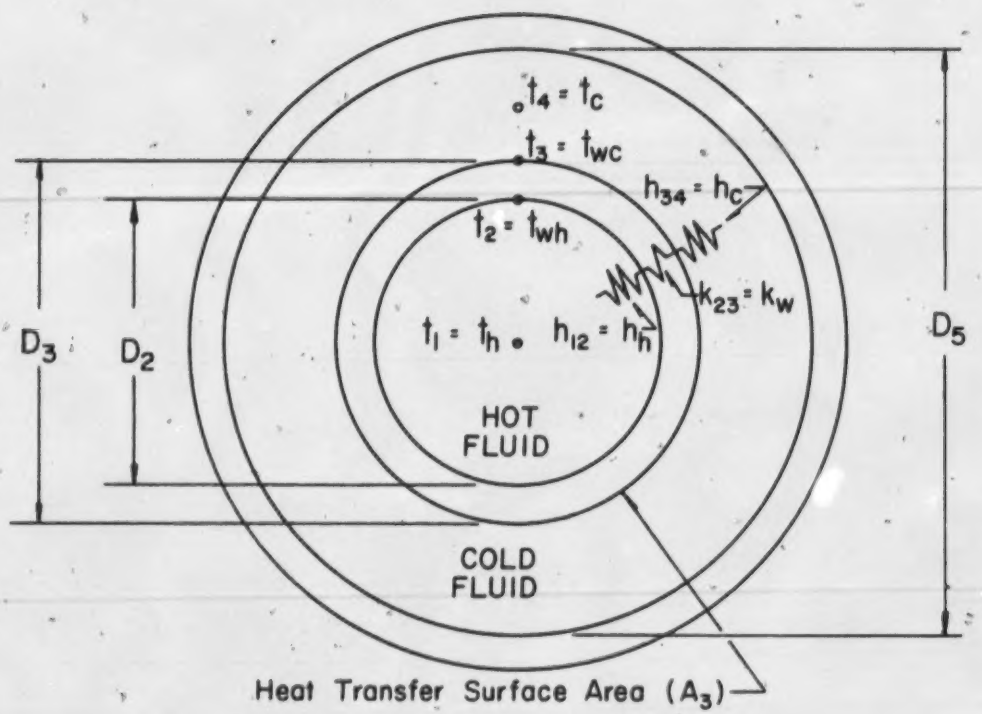


Fig. 4 CROSS SECTION OF A SINGLE PASS HEAT EXCHANGER

If scale develops on the walls, this will form additional resistances.

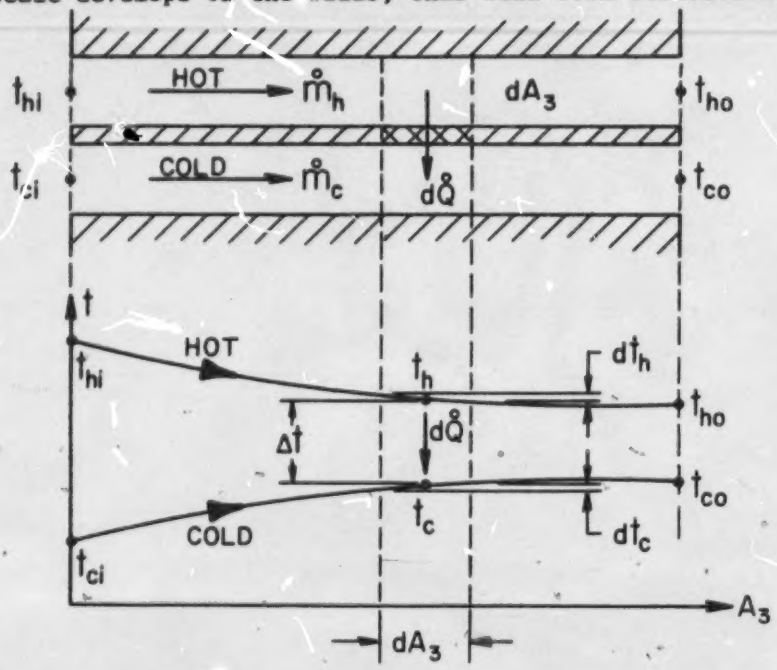


Fig. 5 PARALLEL FLOW HEAT EXCHANGER

In Figure 5, the fluid temperatures are plotted against the heat transfer area, A_3. If we look at some small increment of area, dA_3, there will be some temperature difference, Δt causing an increment heat transfer rate, $d\dot{Q}$ related as follows:

$$d\dot{Q} = U_3 \, dA_3 \, \Delta t \qquad (3)$$

Both U_3 and Δt are local values, which vary continuously over the length of the exchanger. Calculations will be much simpler if we can replace these local values with mean values for the total exchange area. The mean temperature difference for this type of exchanger is called the log mean temperature difference.

$$\Delta t_{lm} = \frac{\Delta t_1 - \Delta t_2}{\ln \frac{\Delta t_1}{\Delta t_2}} \qquad (4)$$

The temperature differences, Δt_1 and Δt_2 are calculated differently for parallel flow and counterflow conditions, as may be seen from Figure 1 and 2. We have now, for the overall rate of heat transfer.

$$\dot{Q} = U_m A_3 \Delta t_{lm}, \qquad (5)$$

where	$\dot{Q}$ = rate of heat transfer	Btu/hr
	U_m = overall mean heat transfer coefficient	Btu/hr-ft^2-^{o}F
	A_3 = transfer area	ft^2
	Δt_{lm} = log mean temperature difference	^{o}F.

For exchangers of more complex design, Δt_{lm} must be modified with an experimentally determined correction factor.

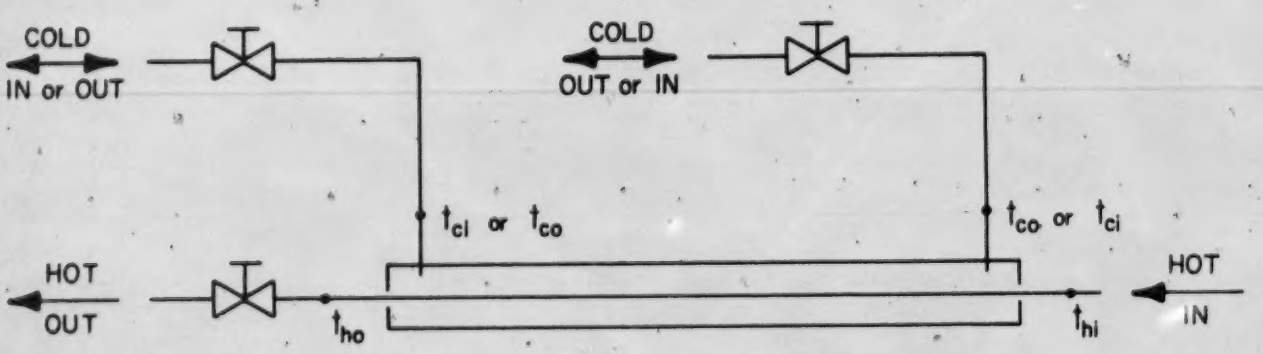


Fig. 6 SINGLE PASS HEAT EXCHANGER

Figure 6 is a diagram of the heat exchanger used in this experiment. If we measure the inlet, and outlet temperatures, we can calculate the log mean temperature differences (see Figures 1 and 2 for Δt_1 and Δt_2). If we measure the rate of flow of hot water we can calculate the heat transfer rate from:

$$\dot{Q} = \dot{m}_h \, C_{ph} \, \Delta t_h \tag{6}$$

where $\dot{m}_h$ = mass rate of flow of hot water lbm/hr

C_{ph} = specific heat at constant pressure of the hot water Btu/lbm-^{o}F

The rate of heating of the cold fluid is equal to the rate of cooling of the hot fluid. For the cold side we have

$$\dot{Q} = \dot{m}_c \, C_{p_c} \, \Delta t_c \tag{7}$$

from which we can obtain the rate of flow of cold water, $\dot{m}_c$. With the calculated valves of $\dot{Q}$ and Δt_{lm}, we can compute the overall heat transfer coefficient, U_m from equation (5).

If we assume turbulent flow of hot water ($Re_{D_2} > 2300$), we have, from McAdams,

$$Nu_{D_2} = .023 \, Re_{D_2}^{.8} \, Pr^{.3} \tag{8}$$

where Nu_{D_2} = Nusselt number based on D_2

Re_{D_2} = Reynolds number based on D_2

Pr = Prandtl number

The Prandtl number, and the properties used to evaluate the Reynolds and Nusselt numbers depend on the fluid temperature, and are usually based on the average fluid temperature, t_{ha}.

The film coefficient, h_{12} on the inside wall of the hot water tube may be calculated from the Nusselt number,

$$Nu_{D_2} = \frac{h_{12} D_2}{k_1} \tag{9}$$

where h_{12} = film coefficient on the hot side — Btu/hr-ft^2-^{o}F

D_2 = hot water tube inside diameter — ft

k_1 = thermal conductivity of hot water at average hot water temperature, t_{ha} — Btu/hr-ft-^{o}F

As with all thermal devices, we like to know the effectiveness of the heat exchanger, and compare the effectiveness in the parallel flow mode with that in the counter flow mode. The heat exchanger effectiveness is defined as the ratio of the actual heat transfer rate to the max. rate obtained from a counterflow exchanger of infinite length with the same inlet temperatures.

$$\varepsilon = \frac{\dot{Q}\ \text{actual}}{\dot{Q}\ \text{max}} \tag{10}$$

The heat transfer rate of an infinite counterflow exchanger depends on which side of the exchanger has the larger heat capacity rate, $\dot{m}\ C_p$. If

$$\dot{m}_h\ C_{ph} < \dot{m}_c\ C_{pc}$$

the hot fluid cools faster than the cold fluid heats, and if the exchanger is long enough, the hot fluid will cool to the inlet temperature of the cold fluid, $t_{ho} = t_{ci}$.

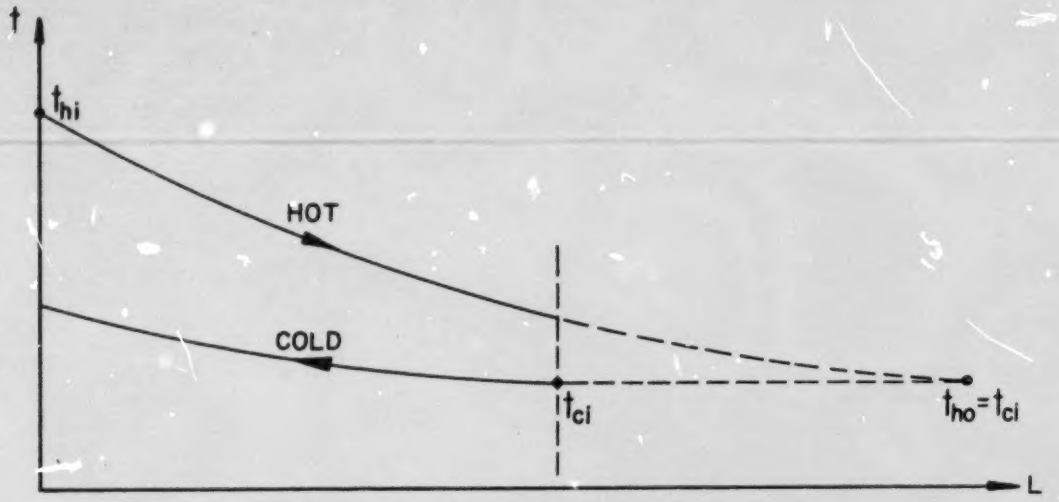


Fig. 7 COUNTER FLOW HEAT EXCHANGER OF INFINITE LENGTH

$$\dot{m}_h C_{ph} < \dot{m}_c C_{pc}$$

The maximum possible heat rate for this condition is

$$\dot{Q}_{max} = \dot{m}_h C_{ph} \Delta t_h$$

$$= \dot{m}_h C_{ph} (t_{hi} - t_{ho})$$

$$= \dot{m}_h C_{ph} (t_{hi} - t_{ci})$$

The exchanger effectiveness is

$$\epsilon = \frac{\dot{m}_h C_{ph} (t_{hi} - t_{ho})}{\dot{m}_h C_{ph} (t_{hi} - t_{ci})}$$

$$\epsilon = \frac{t_{hi} - t_{ho}}{t_{hi} - t_{ci}}, \quad \dot{m}_h C_{ph} < \dot{m}_c C_{pc} \qquad (11)$$

If

$$\dot{m}_h C_{ph} > \dot{m}_c C_{pc},$$

the cold fluid heats faster than the hot fluid cools, and the cold fluid will heat to the hot water inlet temperature, $t_{co} = t_{hi}$, if the exchanger is long enough.

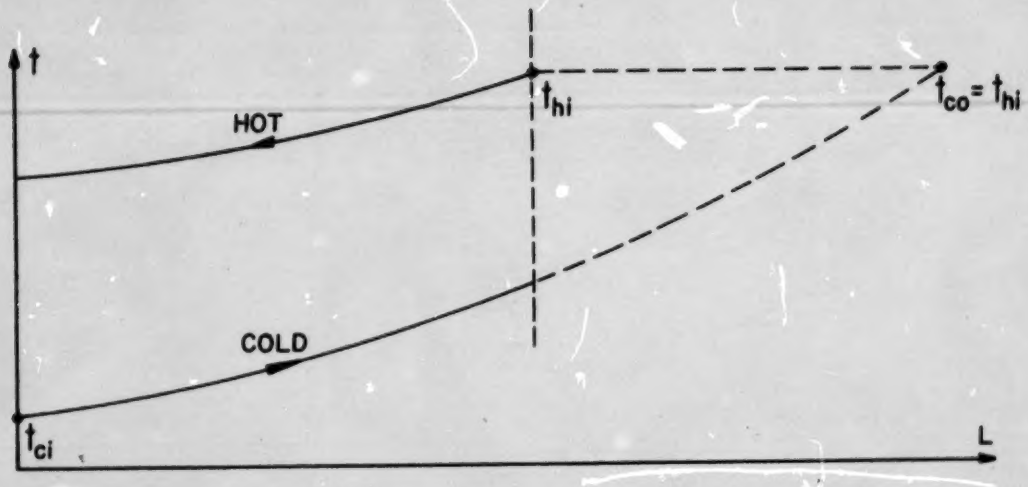


Fig. 8 COUNTER FLOW HEAT EXCHANGER OF INFINITE LENGTH
$\dot{m}_h C_{ph} > \dot{m}_c C_{pc}$

Now the maximum possible heat transfer rate is

$$\dot{Q}_{max} = \dot{m}_c C_{pc} \Delta t_c$$

$$= \dot{m}_c C_{pc} (t_{co} - t_{ci})$$

$$= \dot{m}_c C_{pc} (t_{hi} - t_{ci}),$$

and the effectiveness is

$$\varepsilon = \frac{t_{co} - t_{ci}}{t_{hi} - t_{ci}}, \qquad \dot{m}_h C_{ph} > \dot{m}_c C_{pc} \tag{12}$$

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HEAT EXCHANGER EXPERIMENT

OBJECT

This experiment was designed to serve as a further study of the principles of convective heat transfer, and their application to heat exchangers.

From the analysis of the data may be obtained values for the overall heat transfer coefficient, the film coefficients on both sides of the transfer wall, and the effectiveness of the heat exchanger in both modes of operation.

APPARATUS

A single pass heat exchanger is employed, it can be run both as a parallel flow and as a counter flow device. The hot water runs in the inner tube, surrounded by cold water. Four mercury thermometers measure inlet and outlet temperatures, and the hot water flow rate is determined with the aid of a stop watch and weighing tank.

PROCEDURE

To perform the calculations as outlined in the introduction, we need to measure the inlet and outlet temperatures of the hot and cold water, and the mass rate of flow of hot water. The mass rate of flow is obtained from a measurement of the time required to collect a given amount of water.

The cold water flow is kept constant while the hot water is varied over 7 different rates by adjusting the hot water needle valve. The following procedure may produce the desired data:

1. Open both the hot and cold water valves to flush the lines.
2. Connect the cold water hoses for parellel flow.
3. Open the cold water inlet valve fully.
4. Open the cold water outlet valve 2 full turns.

5. Connect the hot water hose and open the hot water valve as indicated on the data sheet.
6. Open the drain in the collecting tank.
7. Let the water run until the temperatures remain constant.
8. Adjust the balance to be a few pounds over the weight of the tank.
9. Close the tank drain.
10. Start stopwatch at moment the balance beam moves up.
11. Record temperatures; add 30 lbs. to the balance (beam will move down again).
12. Stop time when beam moves up.
13. Record temperatures and time.
14. Open tank drain.
15. Reset hot water needle valve.
16. Repeat procedure from item 7 on.
17. Plot time in secs vs hot water valve setting to check flow data.
18. After 7 runs in parallel mode, interchange hoses on cold water side for counter flow mode, and repeat procedure from item 3 through item 16.

EVALUATION OF RESULTS

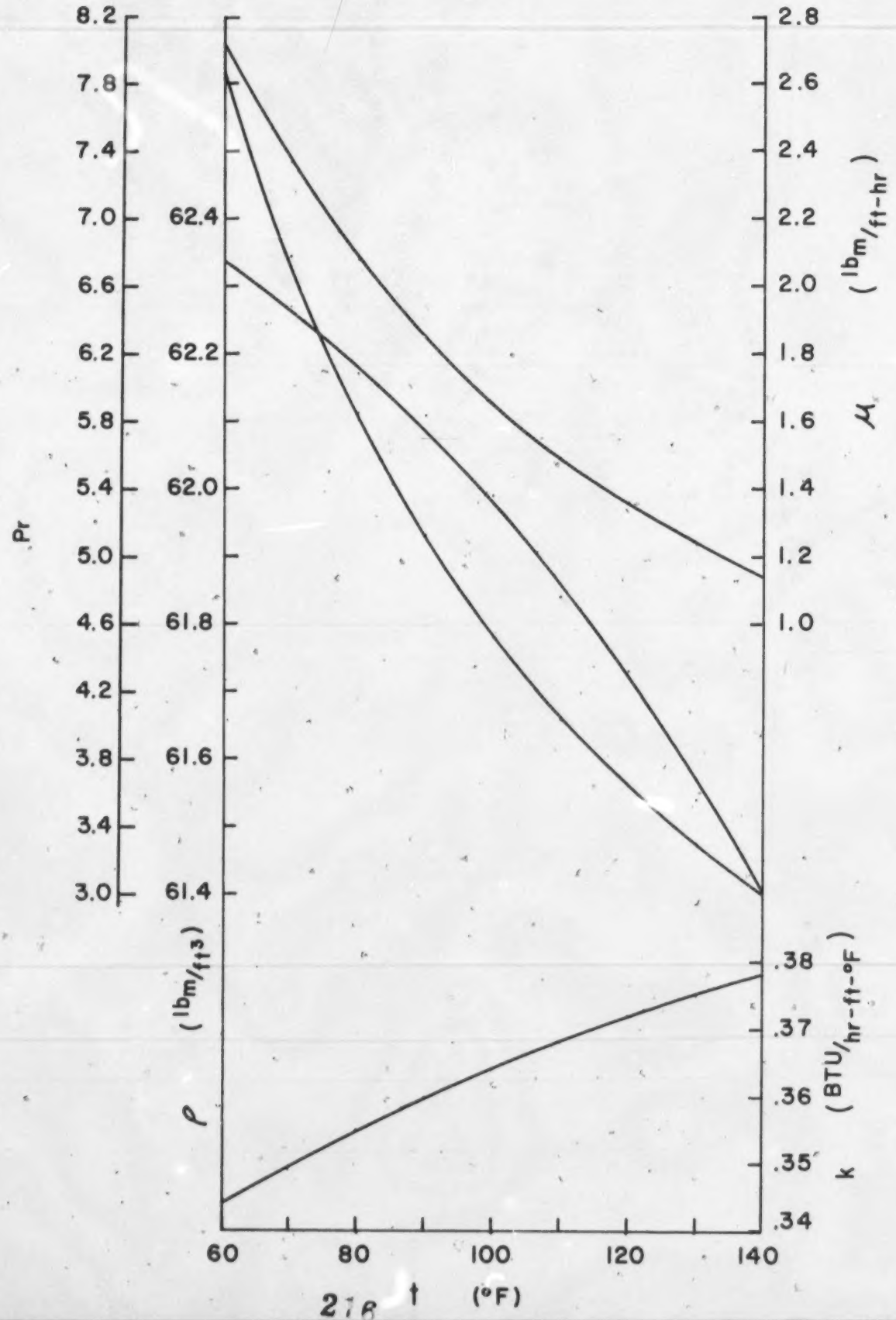
a. Calculate all the quantities indicated in the table of results.

b. Plot the film coefficient, h_{12}, the heat exchanger effectiveness, ε, and the overall mean heat transfer coefficient, U_m vs the mass rate of flow of hot water, $\dot{m}_h$. Plot each quantity for both the parallel flow case and the counterflow case on the same graph.

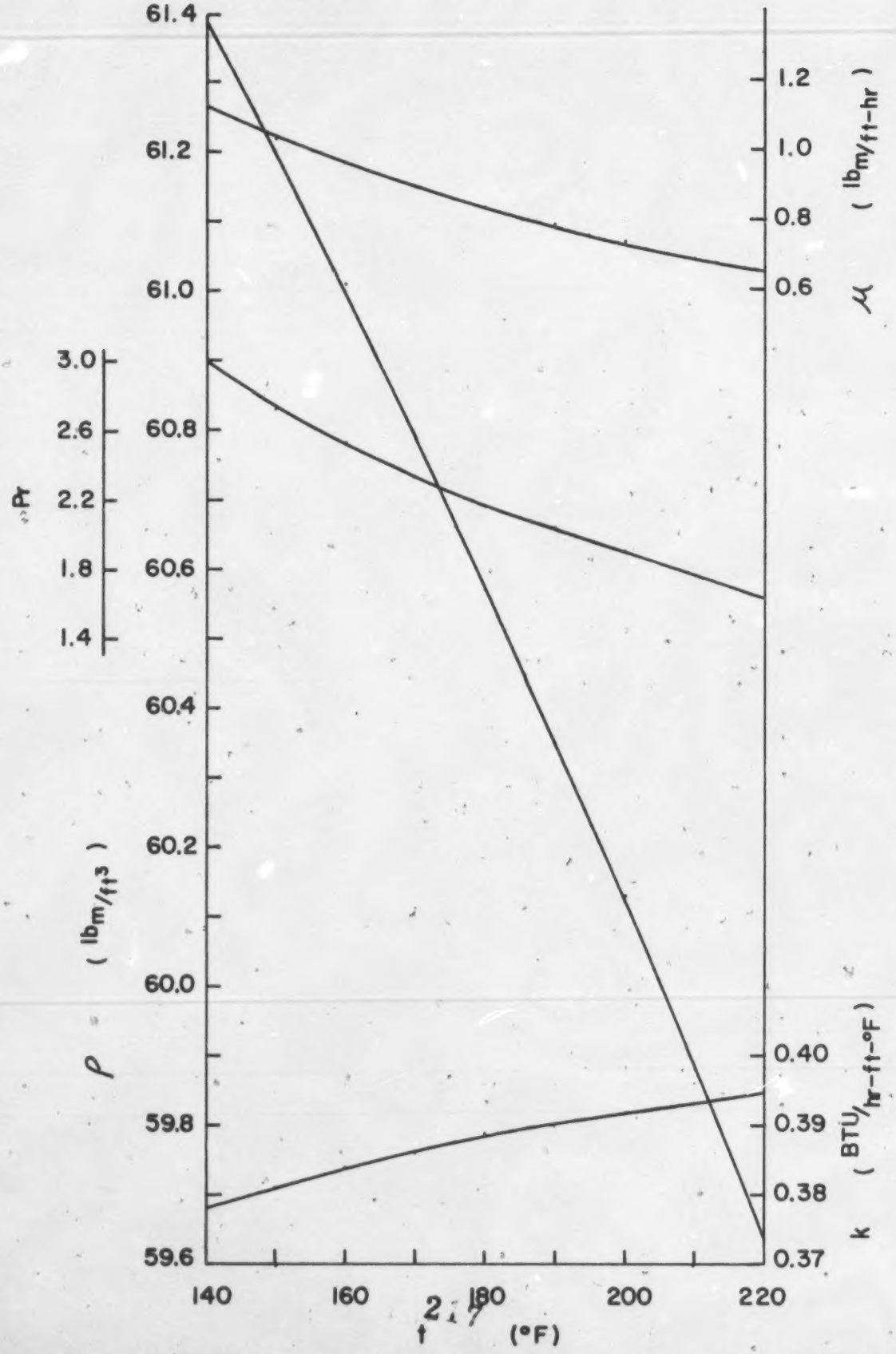
c. What can you say about the relative effectiveness of a counterflow exchanger compared with a parallel flow exchanger?

d. Do you have any comments pertaining to this experiment, and in particular the obtained results? Do you have any suggestions to make this a better experiment?

PROPERTIES OF WATER (60 to 140 °F)



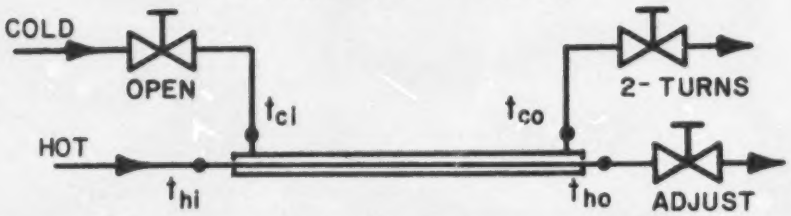
PROPERTIES OF WATER (140 to 220 °F)

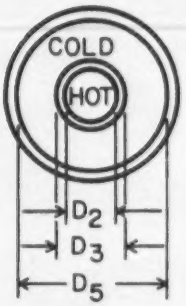


DATA

EFFECTIVE LENGTH = 25" D_2 = .305 D_3 = .375 D_5 = .545

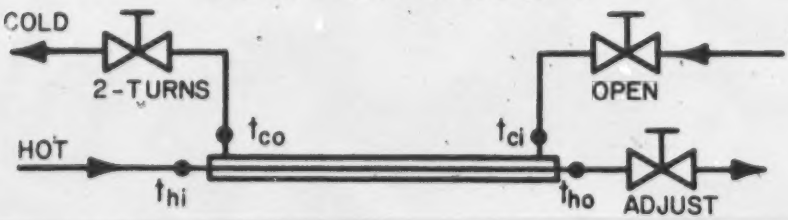
PARALLEL FLOW





HOT WATER VALVE (TURNS)	TEMPERATURES (°F)				TIME (MIN.-SEC.)	WEIGHT OF HOT WATER (LB_F) *
	HOT WATER		COLD WATER			
	t_{hi}	t_{ho}	t_{ci}	t_{co}		
1						
1-1/4						
1-1/2						
1-3/4						
2						
2-1/4						
2-1/2						

COUNTER FLOW



* SUGGESTION: COLLECT 30 lb_f IN EACH RUN

1						
1-1/4						
1-1/2						
1-3/4						
2						
2-1/4						
2-1/2						

RESULTS

PARALLEL FLOW										
$\dot{m}$ (lb_m/hr)	t_{ha} (°F)	t_{ca} (°F)	Re @ D_2	Nu @ D_2	Δt_{lm} (°F)	Um $\left(\frac{BTU}{hr\text{-}ft^2\text{-}°F}\right)$	$h_{1\text{-}2}$ (BTU/lb_m)	$h_{3\text{-}4}$ (BTU/lb_m)	CR	ϵ
COUNTER FLOW										

Conduction heat transfer deals with the flow of thermal energy, called heat, through a conducting medium. The quantity of heat so transferred per unit time over a unit distance depends on the following parameters:

1. The temperature gradient in the conductor.
2. The thermal conductivity of the conductor.
3. The cross sectional area of the conductor normal to the direction of heat flow.

The temperature gradient, or slope at each point along the path of conduction is the limiting value of the ratio $t_2 - t_1 / x_2 - x_1$, as illustrated below. It is negative in the direction of heat flow.

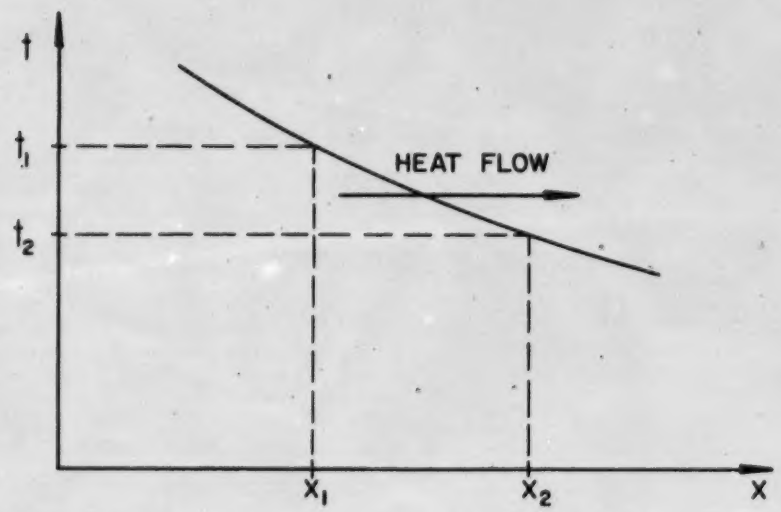


Fig. I HEAT CONDUCTION

The thermal conductivity is a property of the material of the conductor, and it indicates its relative resistance to heat flow.

Mathematically the relationship may be expressed by what is known as the Fourier equation for conduction:

$$\frac{dQ}{d\tau} = kA \frac{\partial t}{\partial x} \qquad (1)$$

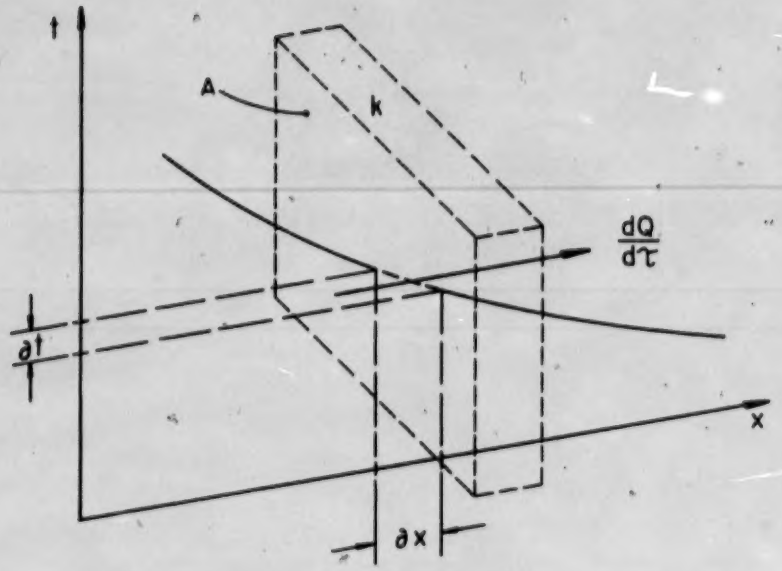


Fig. 2 FOURIER HEAT CONDUCTION EQUATION

dQ = the heat transferred in time dt over a distance ∂x	Btu
dτ = the time	hr
∂t = the change in temperature over the distance ∂x	^{o}F
∂x = the length of the path of conduction	ft
A = the cross sectional area of the conductor normal to the path	ft^2
k = the thermal conductivity	Btu/hr ft ^{o}F

In the general, non steady case, where the process changes with time, the temperature along the path of conduction depends not only on the distance, x along the path, but also on the time, τ. It is for this reason that the temperature gradient is represented by the partial derivative, $\frac{\partial t}{\partial x}$.

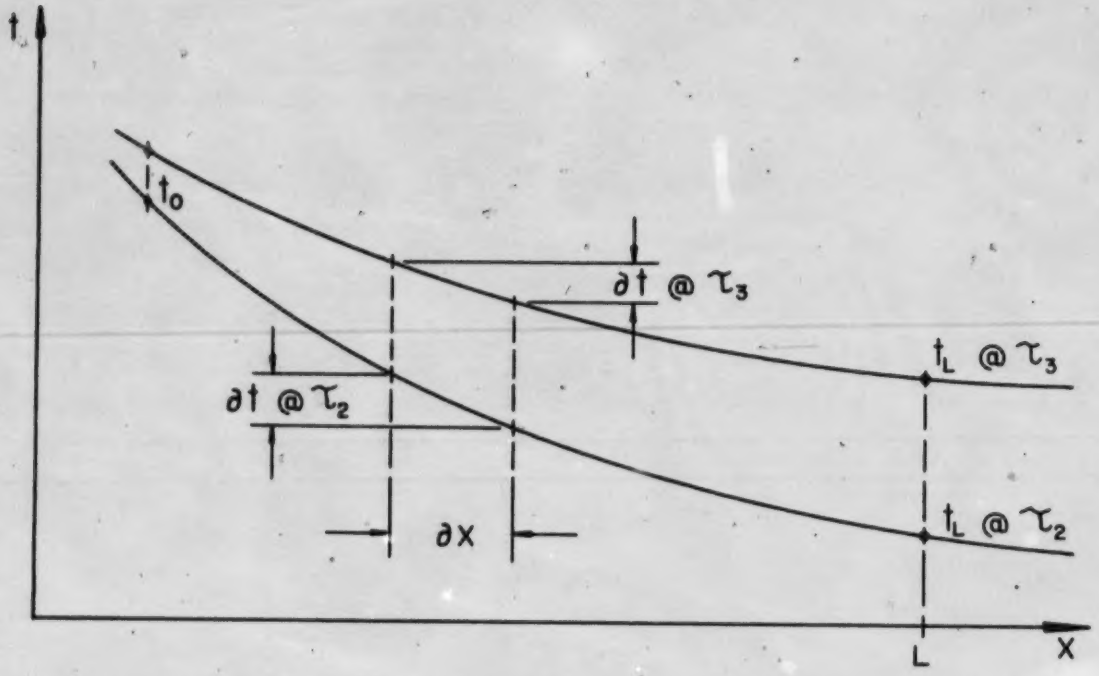


Fig. 3 TIME DEPENDENT TEMPERATURE GRADIENT

REFERENCES

Chapman, Alan J., Heat Transfer, New York, The MacMillan Co., 1960.

Kreith, Frank, Principles of Heat Transfer, Scranton, International Textbook Co., 1963.

Schneider, P.J., Conduction Heat Transfer, Reading, Addison-Wesley Publishing Co., INc., 1957.

METALS AS HEAT CONDUCTORS EXPERIMENT

OBJECT

This experiment demonstrates the wide range of thermal conductivities found in metals, and their corresponding effectiveness as heat conductors. A heat pipe is included to demonstrate its great heat transport capacity, within a certain temperature range.

The second part of the experiment is meant as an exercise in data analysis. The data obtained in the first part is further analyzed to extract an estimate of thermal conductivities of the various metals.

APPARATUS

Three metal rods and a heatpipe form heat conductors between hot and cold reservoirs, embedded in thermal insulation. Four mercury thermometers, eight thermocouples, a potentiometer, and a stop watch are required to obtain the necessary data. The rods and heat pipe each have two thermocouples, equally spaced. The thermocouples in each rod are connected as shown in Figure 4 to obtain the potential difference between the two thermocouples. The conversion of this difference in mV to ^{o}F represents the temperature difference $\Delta t = t_b - t_c$ between the two points in the rods.*

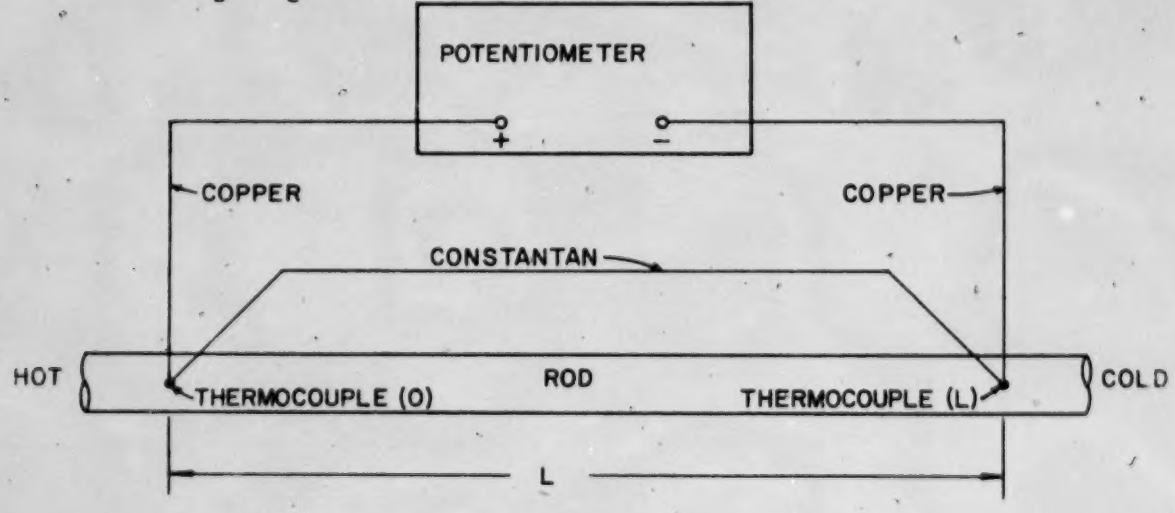
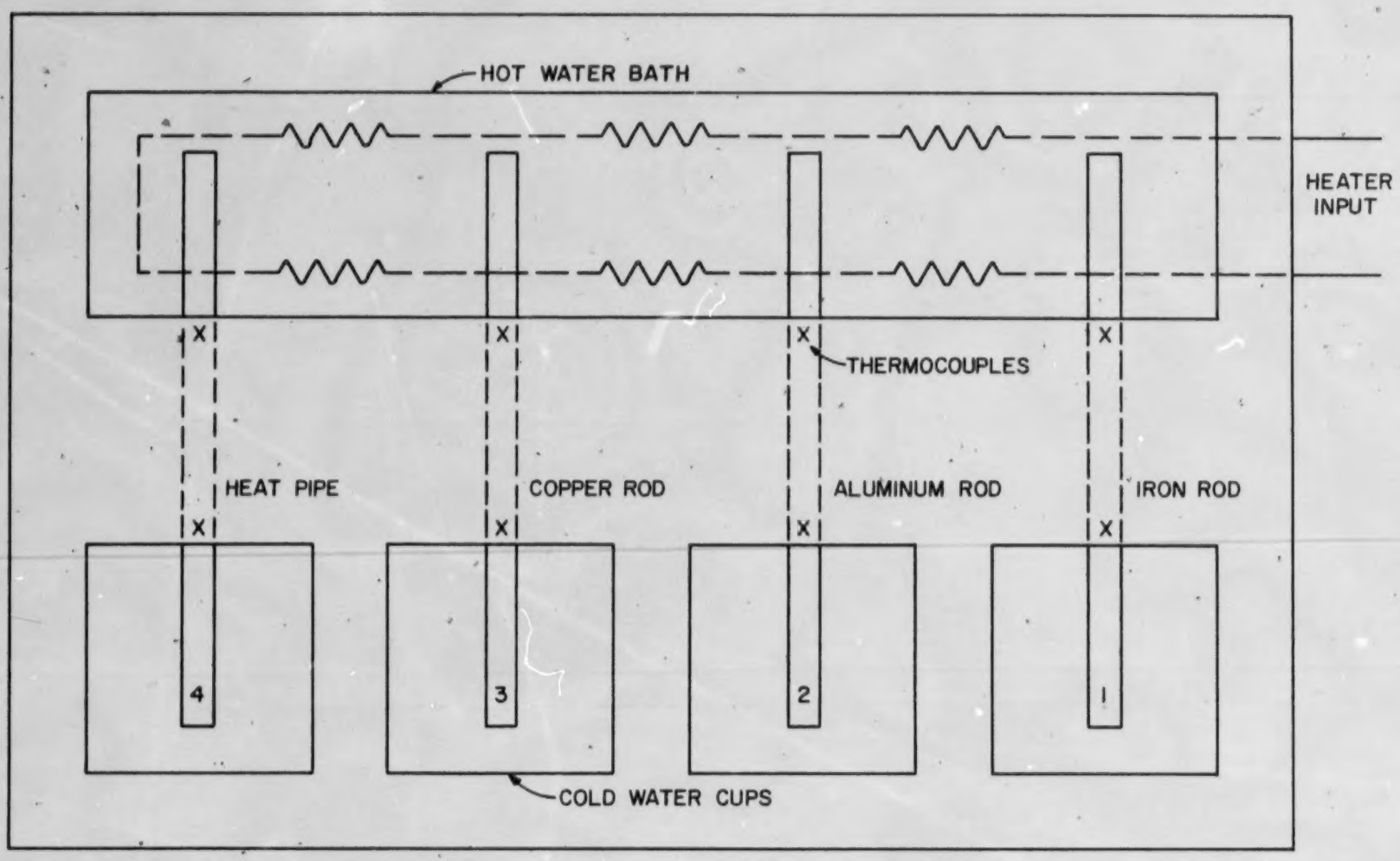
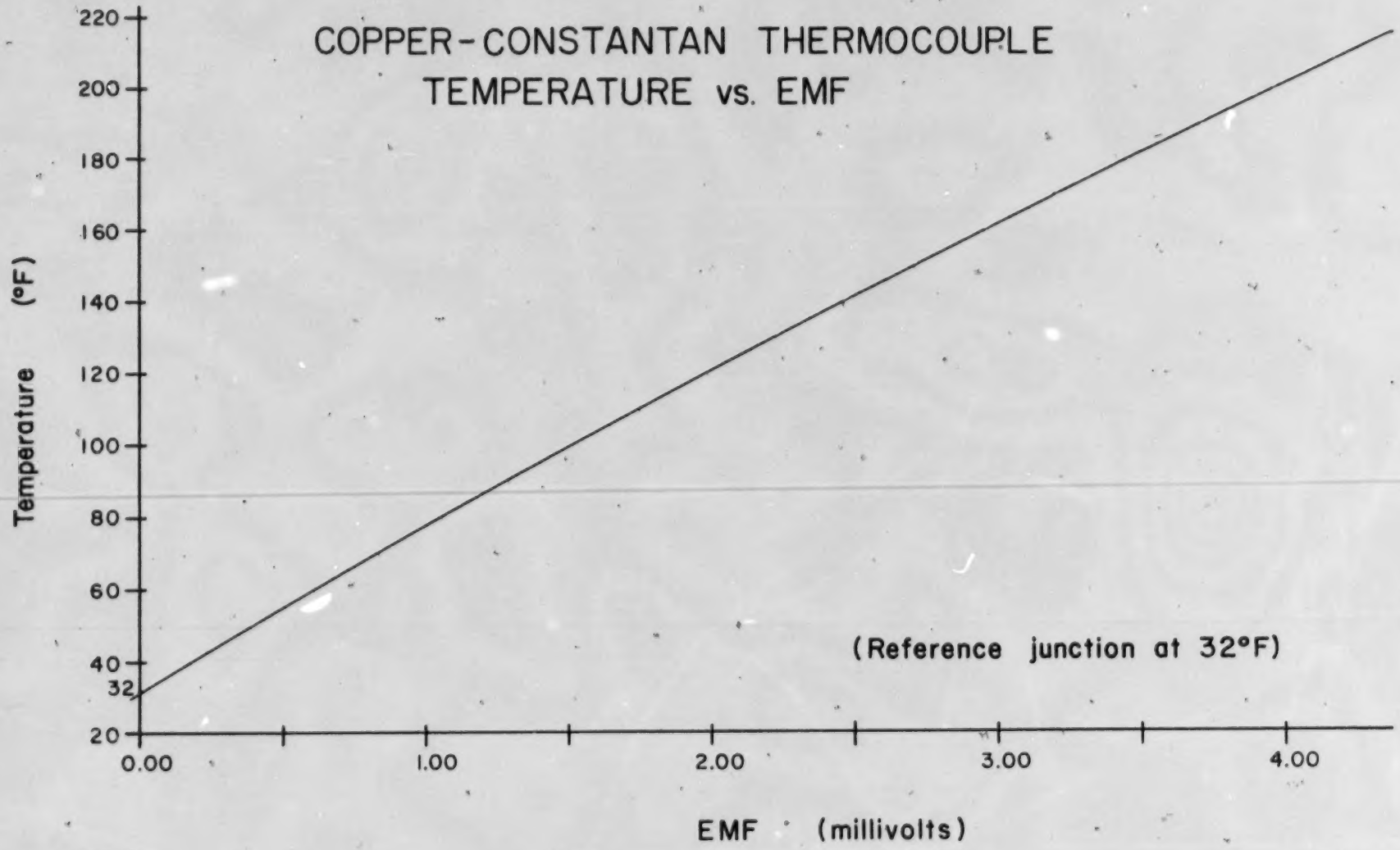


Fig. 4 THERMOCOUPLE HOOKUP

* The emf of a thermocouple is not a linear function of its temperature. The ΔE is mV must be converted to ^{o}F at the appropriate place in the table. (See the graph on next page).



CONDUCTION HEAT TRANSFER APPARATUS


COPPER-CONSTANTAN THERMOCOUPLE
TEMPERATURE vs. EMF
Temperature (°F)
220
200
180
160
140
120
100
80
60
40
32
20
0.00
1.00
2.00
3.00
4.00
(Reference junction at 32°F)
EMF (millivolts)

PROCEDURE

Fill the hot water bath of the apparatus to the level indicated. Poor cold water and two ice cubes in a beaker to the 360 ml mark, weigh and poor in a cold water cup. Weigh and record the emptied beaker - repeat for all cups. Turn on the heater full power until the water begins to boil, then turn down to just maintain boiling.

This experiment requires three persons - one to read time and record data, one to read the potentiometer, and one to read the thermometers. Data taking may be done as follows: The time keeper indicates every 30 sec. interval at which the potential difference for one rod, and the temperature of the corresponding cup is read and recorded. At the next time interval the next rod and cup values are recorded, etc. This procedure is continued until all temperatures become steady. The length of the time interval may of course be changed to suit the experimenters. The water in the individual cups must be stirred slowly to insure proper temperature readings.

EVALUATION OF RESULTS

Convert all ΔE readings to the temperature differences Δt in the rods, and compute the mass of water in lbm in each cup.

To obtain a better understanding of the mechanics of heat conduction, the variables involved are plotted in various relationships.

Part-A

For each water temperature the heat input may be calculated from:

$$Q_w = M_w C_w (t_{w\tau} - t_{w\tau_o}) \qquad (2)$$

where

Q_w	= heat added to water	Btu
M_w	= mass of water	lbm
C_w	= specific heat of water	Btu/lbm^o

$t_{w\tau}$ = temperature of water at time = τ ^{o}F

$t_{w\tau_o}$ = temperature of water at time = τ_o ^{o}F

Item-A1

Plot the water temperatures $t_{w\tau}$ against time τ. Plot all four cups in the same graph and indicate a time τ_o for each cup at some data point where the plots have a slope greater than zero.

Item-A2

At each datum point after τ_o compute the heat input to each cup from equation (2), and plot Q_w against the time $(\tau - \tau_o)$ four curves in same graph).

Item-A3

For each interval between data points compute the rate of heat input to the water in each cup from:

$$\frac{\Delta Q_w}{\Delta \tau} = \frac{Q_{w\,n} - Q_{w\,n-1}}{\Delta \tau} \qquad (3)$$

where $\Delta\tau = \tau_n - \tau_{n-1}$, time interval between two consecutive data points for the same cup. Plot the heat rate $\frac{\Delta Q_w}{\Delta \tau}$ against time $(\tau - \tau_o)$ for all four cups in one graph.

Item-A4

Plot the temperature differences Δt for all four rods against the time, $(\tau - \tau_o)$ in one graph.

Item-A5

From the information in items - A-3 and A-4 construct plots of the heat rates $\frac{\Delta Q_w}{\Delta \tau}$ against the temperature differences, Δt at cor-

responding times $(\tau-\tau_o)$.

Part B

From equation (1) we obtain

$$dQ = kA \frac{\partial t}{\partial x} d\tau$$

The average of the partial derivative, $\frac{\partial t}{\partial x}$ over the path length, L at any time, τ may be roughly approximated by the ratio $-\frac{\Delta t}{L}$ at the same time, τ (see Figure 5),

$$dQ \approx kA \frac{\Delta t}{L} d\tau \qquad (4)$$

This means that we are approximating a transient (time dependent) conditio with a series of consecutive steady state conditions. The total heat transferred may then be approximated by integrating equation (4):

$$Q_{0-1} = \int_{dQ \approx kA}^{\tau_1-\tau_o} \int_{\Delta t\ d(\tau - \tau_o)}^{\tau_1-\tau_o} \qquad (5)$$

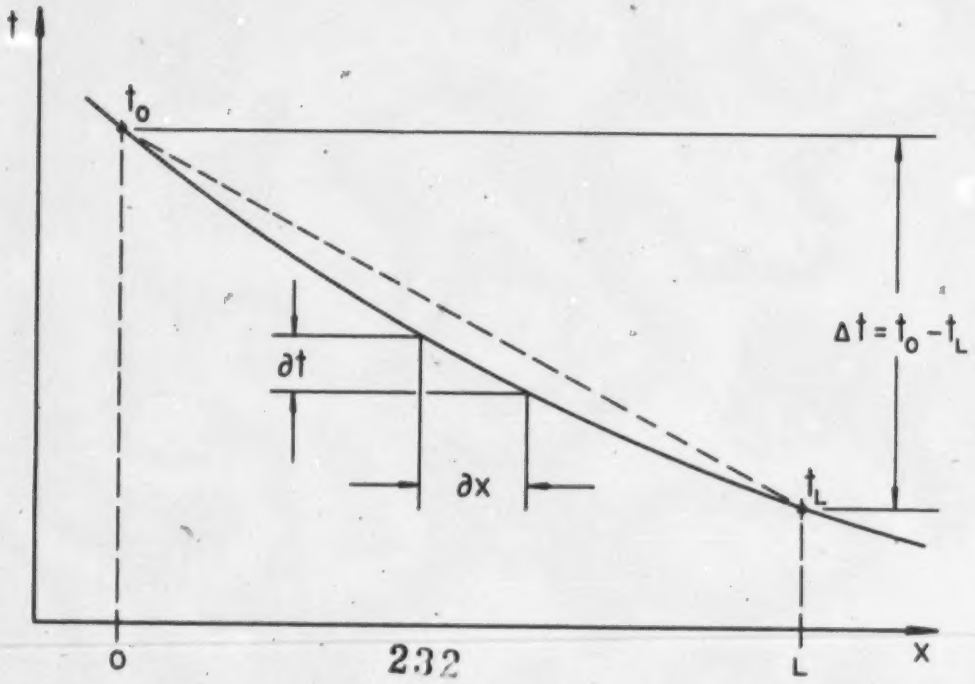


Fig. 5 RELATIONSHIP BETWEEN $\frac{\partial t}{\partial X}$ AND $\frac{\Delta t}{L}$

In item - A-4 we plotted Δt vs the time $(\tau-\tau_o)$. The areas under these curves between zero and $(\tau_1-\tau_o)$ represent the integrals.

$$\int_{\Delta t}^{\tau_1-\tau_o} d(\tau-\tau_o)$$

Thus, we may obtain the values of the integrals by measuring the areas under the curves in the appropriate units. If we assume that all the heat, entering the rods is transferred to the water in the cups, we have, for each rod:

$$Q_{o-1} = Q_{wo-1}$$

where Q_{wo-1} is the heat added to the water between time τ_o and τ_1, and

$$Q_{wo-1} = M_w C_w (t_{w\tau_1} - t_{w\tau_o})$$

From this relationship we may derive an estimate of the thermal conductivity of each rod.

$$k \approx \frac{L\, Q_{wo-1}}{A \int_{\tau_o}^{\tau_1-\tau_o} \Delta t\, d(\tau-\tau_o)} \qquad (6)$$

Item - B-1

Measure the areas under the curves in item - A-4, between times τ_o and τ_1, where τ_1 is a time before the water in the cup has reached its maximum temperature. Convert the areas to the appropriate units. From item - A-2, determine the values of Q_{wo-1} over the same time intervals. Tabulate the corresponding valves for Q_{wo-1}, and the integral.

Item - B-2

Compute the values for the conductivity k from equation (5). From the literature obtain the values for k for the materials used in the experiment. Compute the error in the estimated values in percentage of the literature values. To obtain relative values, divide each value of k by the value for copper. Do this for both the estimated values and those from the literature. Again compute errors in percentage of the literature values. Tabulate all results obtained in this item.

Item - B-3

From the literature describe a method for measuring the thermal conductivity k for a metal. Include a sketch of the experimental arrangement.

Item - B-4

With the aid of a sketch give a brief explanation of the heat pipe.

Item - C

Is there anything about this experiment that, you think, could be improved upon? Is the explanation clear? Is the procedure workable and convenient? Did the apparatus work properly? Are the results meaningful?

DATA

ROD No.1 (IRON)			ROD No. 2 (ALUMN.)			ROD No. 3 (COPPER)			ROD No. 4 (HEAT PIPE)		
Time (sec)	ΔE_1 (mV)	t_{w1} (°F)	Time (sec)	ΔE_2 (mV)	t_{w2} (°F)	Time (sec)	ΔE_3 (mV)	t_{w3} (°F)	Time (sec)	ΔE_4 (mV)	t_{w4} (°F)

RESULTS

No. 1 (IRON)		No. 2 (ALUMN.)		No. 3 (COPPER)		No. 4 (HEAT PIPE)	
MASS OF WATER _____ lb_m		MASS OF WATER _____ lb_m		MASS OF WATER _____ lb_m		MASS OF WATER _____ lb_m	
Time (sec)	Δt_1 (°F)	Time (sec)	Δt_2 (°F)	Time (sec)	Δt_3 (°F)	Time (sec)	Δt_4 (°F)